US009062666B2

(12) **United States Patent**
**Ota et al.**

(10) **Patent No.: US 9,062,666 B2**
(45) **Date of Patent: Jun. 23, 2015**

(54) **SWASH PLATE TYPE VARIABLE DISPLACEMENT COMPRESSOR AND METHOD OF CONTROLLING SOLENOID THEREOF**


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(72) Inventors: **Masaki Ota**, Aichi-ken (JP); **Shohei Fujiwara**, Aichi-ken (JP); **Takahiro Hoshida**, Aichi-ken (JP); **Hiroaki Kayukawa**, Aichi-ken (JP); **Noriyuki Shintoku**, Aichi-ken (JP)

(73) Assignee: **KABUSHIKI KAISHA TOYOTA JIDOSHOKKI**, Aichi-Ken (JP)



( * ) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 309 days.

(21) Appl. No.: **13/742,561**

(22) Filed: **Jan. 16, 2013**

(65) **Prior Publication Data**

US 2013/0189121 A1 Jul. 25, 2013

(30) **Foreign Application Priority Data**

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Mar. 29, 2012 (JP) ................................ 2012-077055

(51) **Int. Cl.**

***F04B 49/02*** (2006.01)
***F04B 1/29*** (2006.01)
***F04B 27/16*** (2006.01)

(52) **U.S. Cl.**

CPC .. ***F04B 1/29*** (2013.01); ***F04B 27/16*** (2013.01)

(58) **Field of Classification Search**

CPC .......... F04B 27/14; F04B 27/16; F04B 27/18; F04B 27/1804
USPC .............................. 417/222.2, 223; 92/70, 71
See application file for complete search history.



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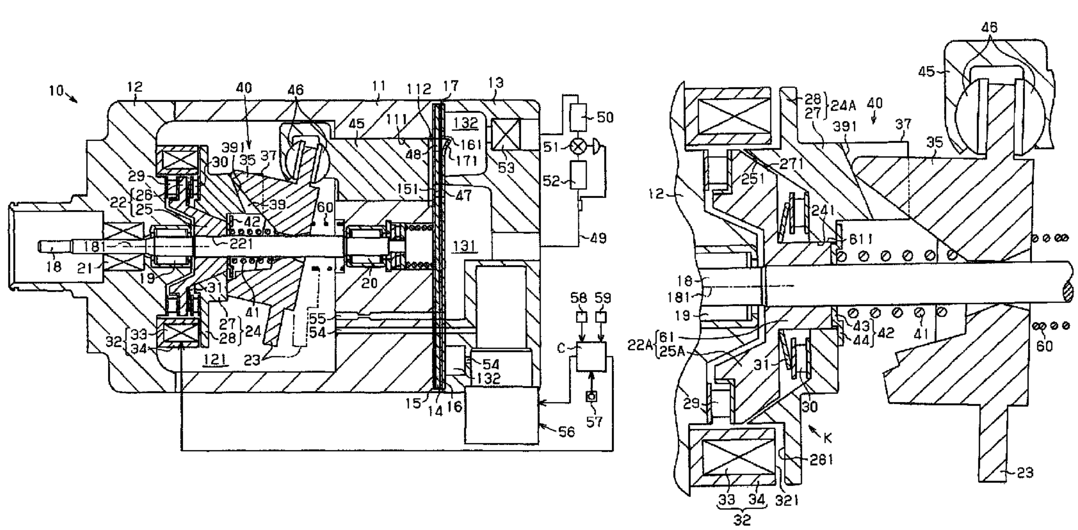
*Primary Examiner* — Charles Freay
(74) *Attorney, Agent, or Firm* — Greenblum & Bernstein, P.L.C.



(57) **ABSTRACT**

The swash plate type variable displacement compressor includes a rotary shaft, a swash plate, a plurality of pistons, a first rotor, a second rotor, a solenoid and a cone clutch. The second rotor transmits the rotation of the first rotor to the swash plate. The solenoid produces electromagnetic force that acts on the first rotor or the second rotor so that the first rotor and the second rotor move toward each other. The cone clutch is engageable by energization of the solenoid. The cone clutch has a male cone portion and a female cone portion. The male cone portion has a conical surface provided on one of the first rotor and the second rotor. The female cone portion has a conical surface provided on the other. The conical surface of the female cone portion is connectable to and disconnectable from the conical surface of the male cone portion.


**32 Claims, 34 Drawing Sheets**



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| JP | 2006-152918 | 6/2006 |
| JP | 2007-24257 | 2/2007 |
| JP | 2008-256210 | 10/2008 |
| JP | 2011-241853 | 12/2011 |

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FIG. 1

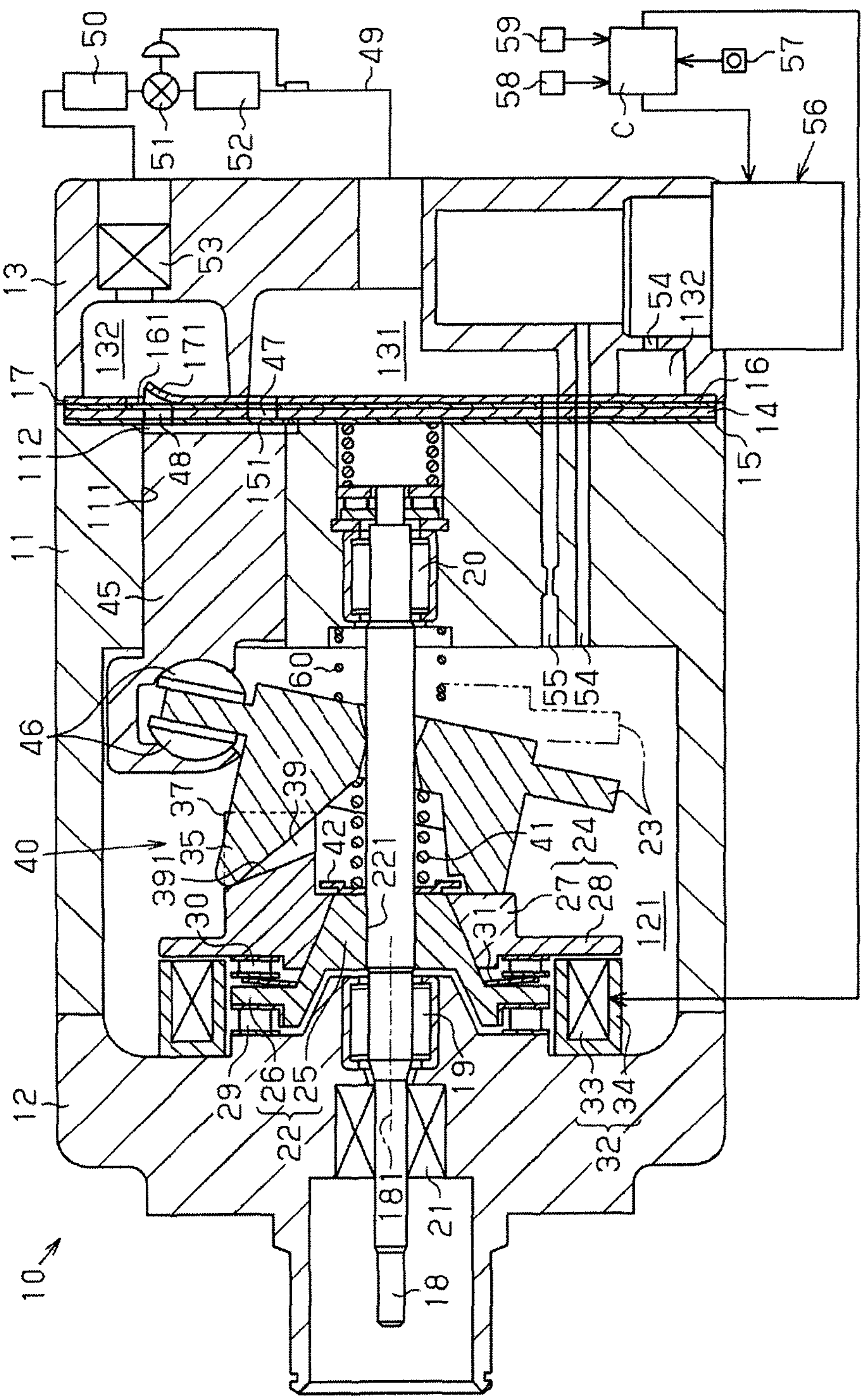
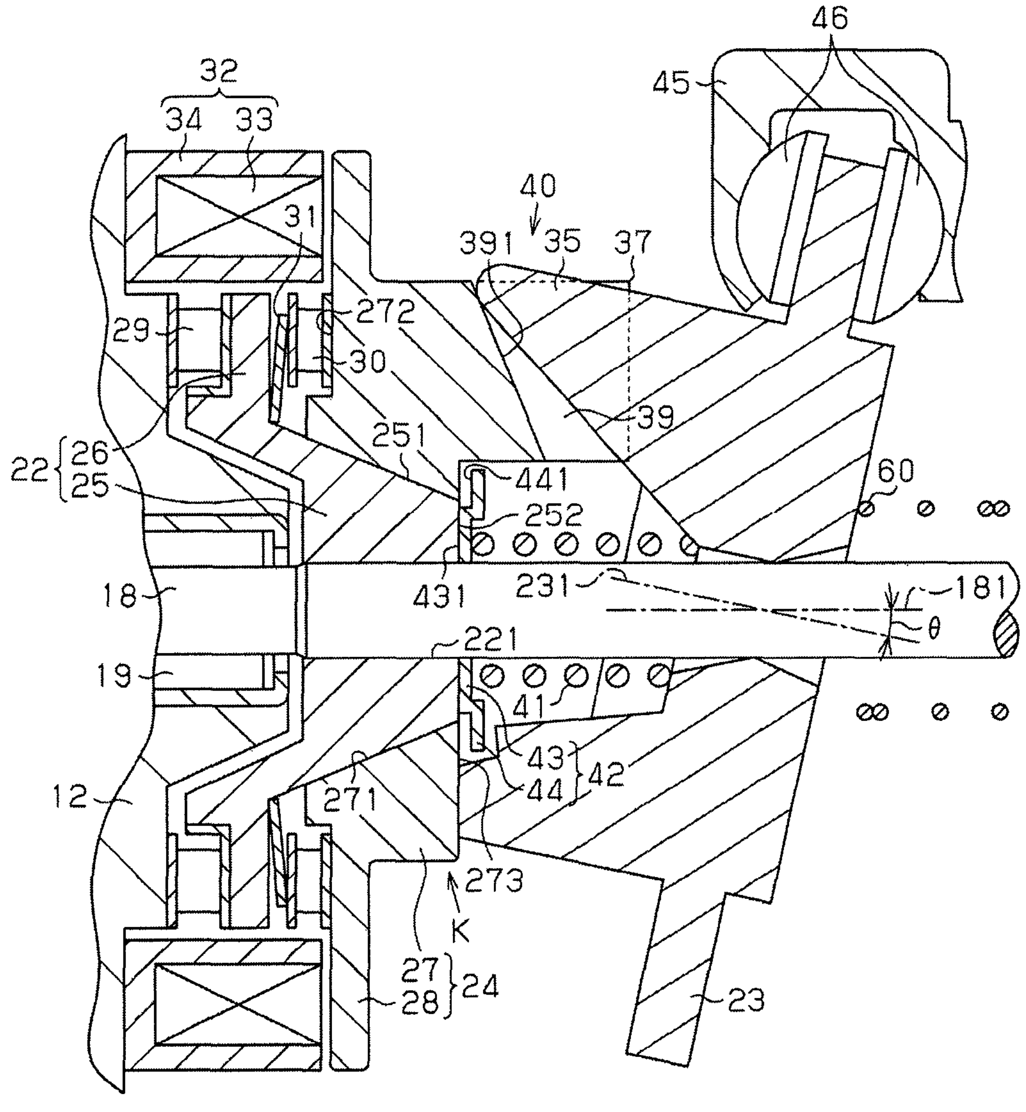


FIG. 2



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391
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431
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θ
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K
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24
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FIG. 3

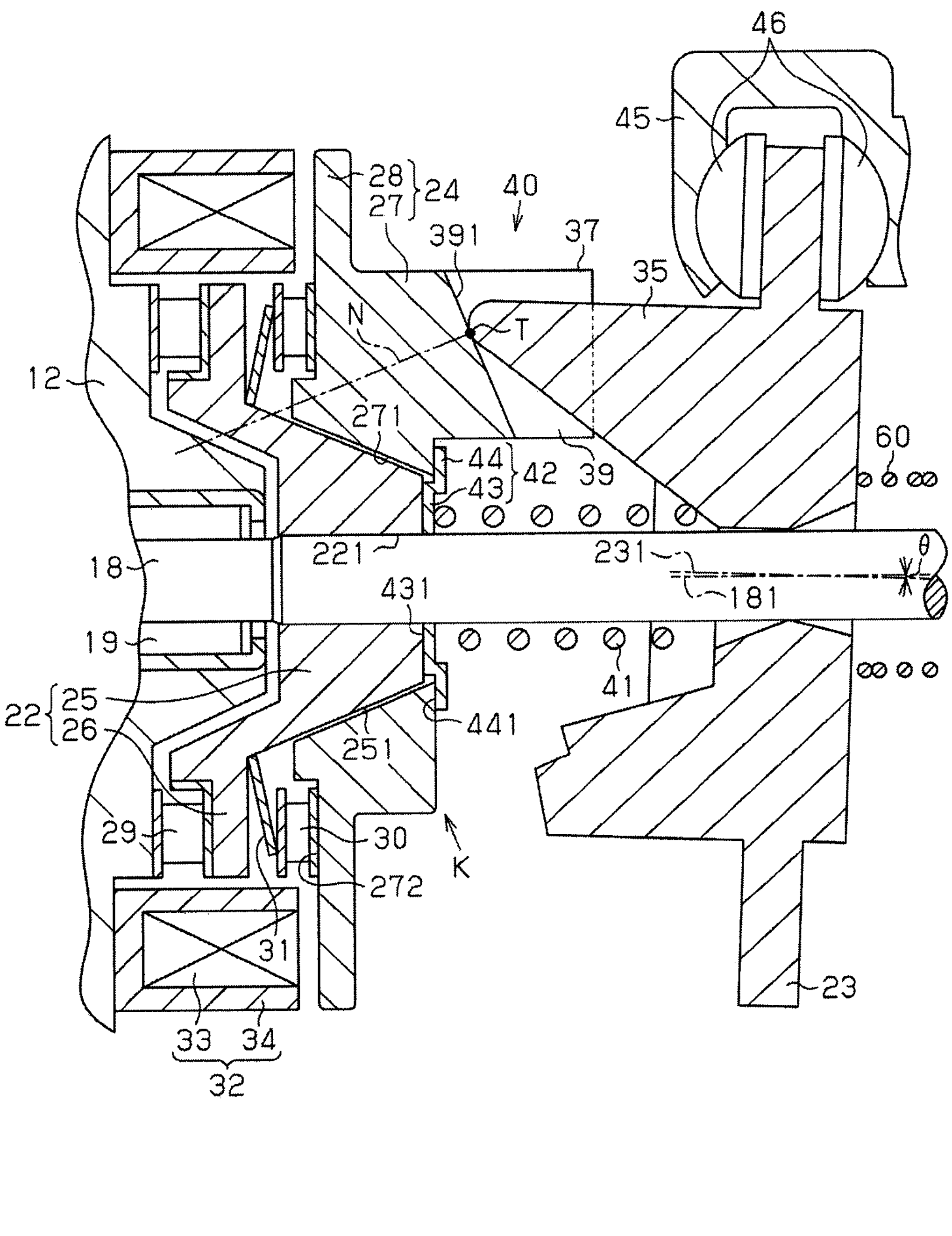
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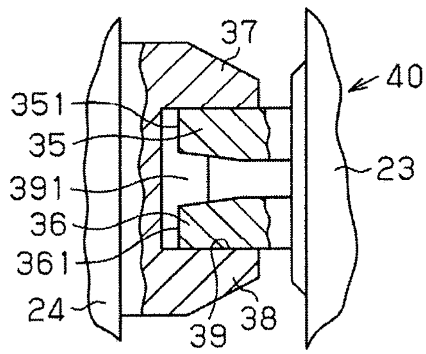
FIG. 4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351
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391
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361
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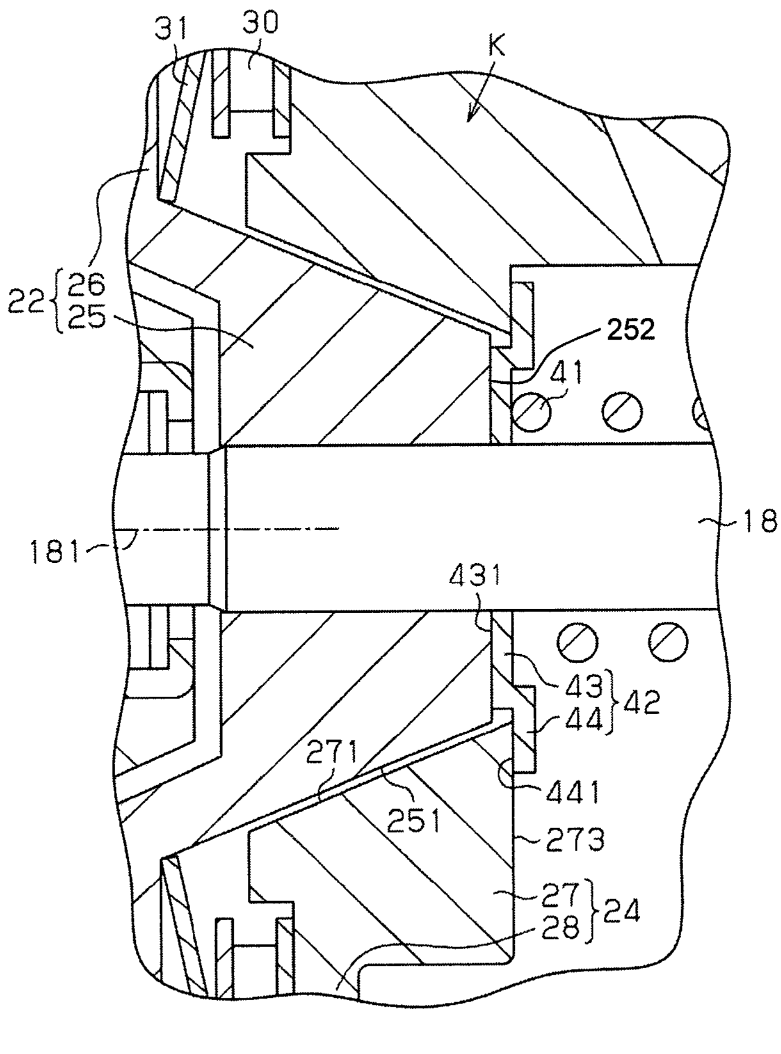
38

FIG. 5


31
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K
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181
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431
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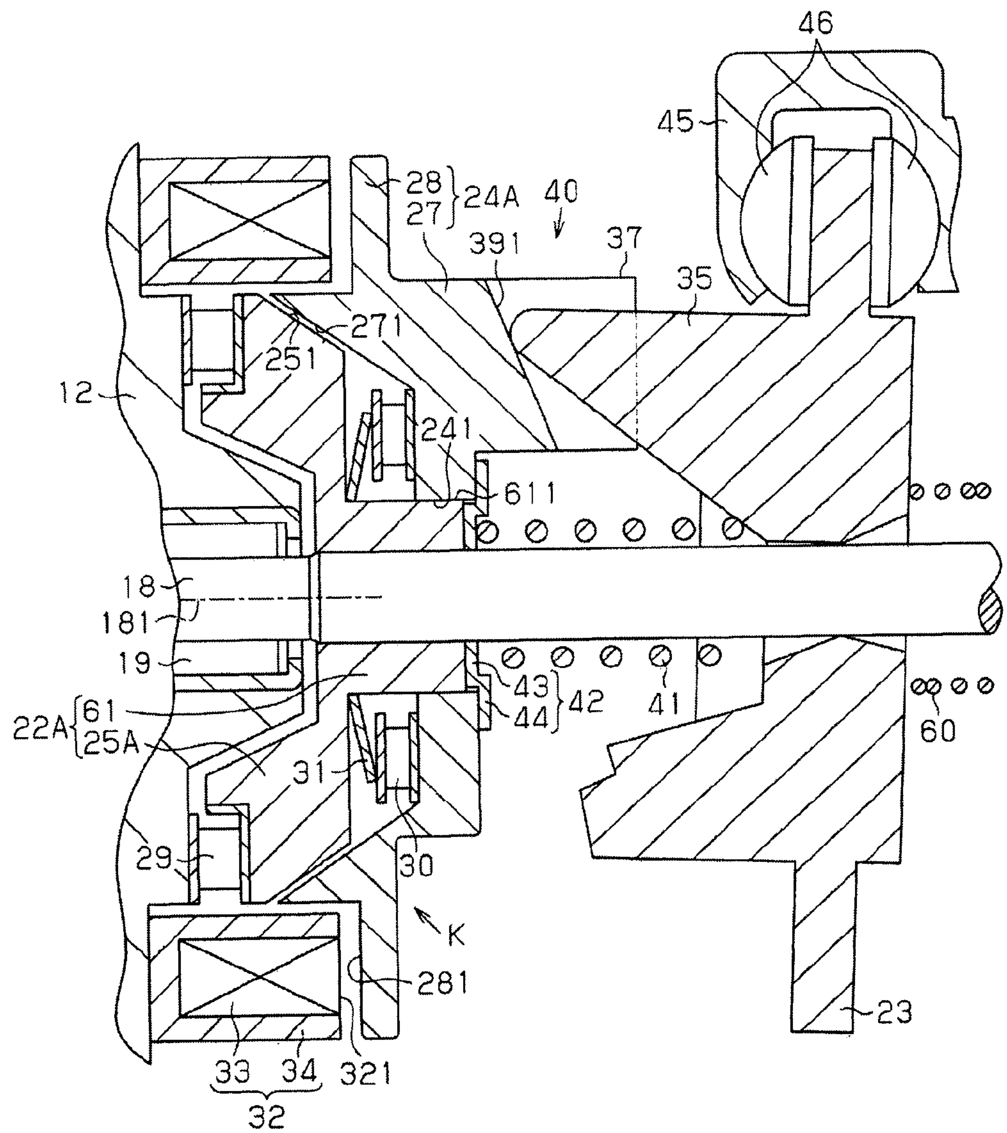
FIG. 6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24A
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391
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271
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181
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22A
61
25A
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44
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41
60
31
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30
K
281
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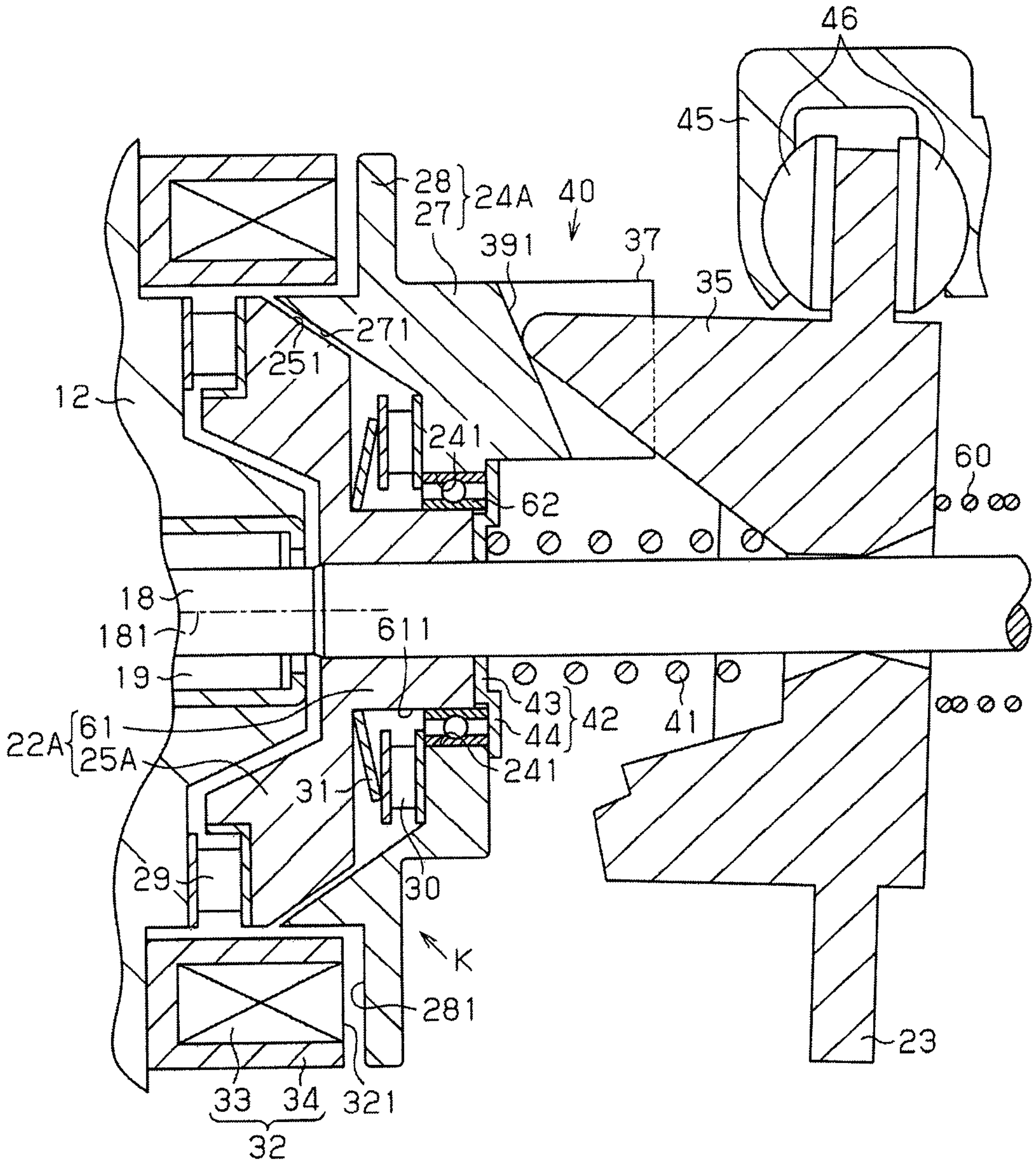
321
32

FIG. 7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24A
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241
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181
611
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22A
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25A
241
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K
281
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321
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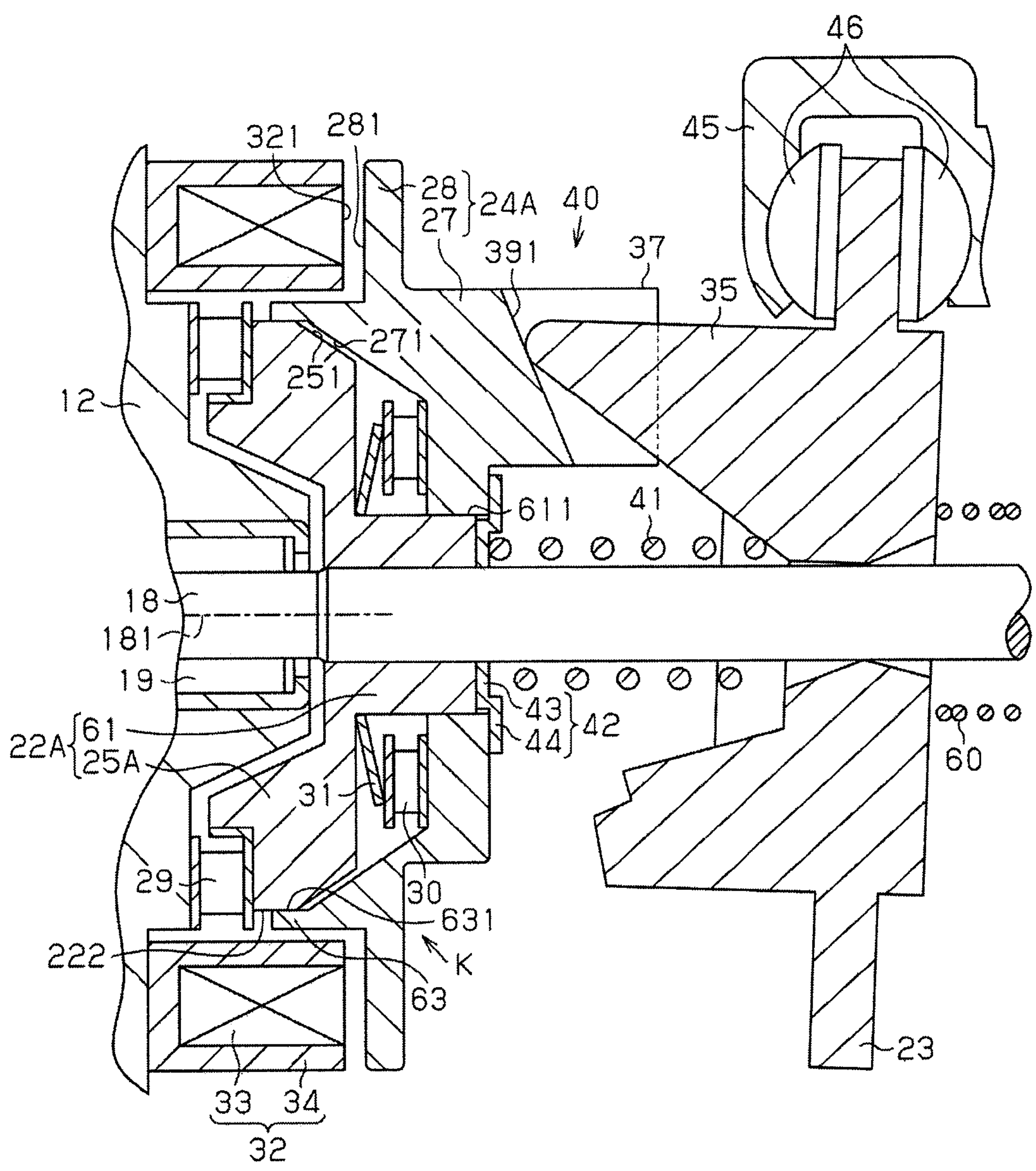
FIG. 8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281
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24A
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391
37
35
271
251
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611
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181
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22A
61
25A
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631
222
K
63
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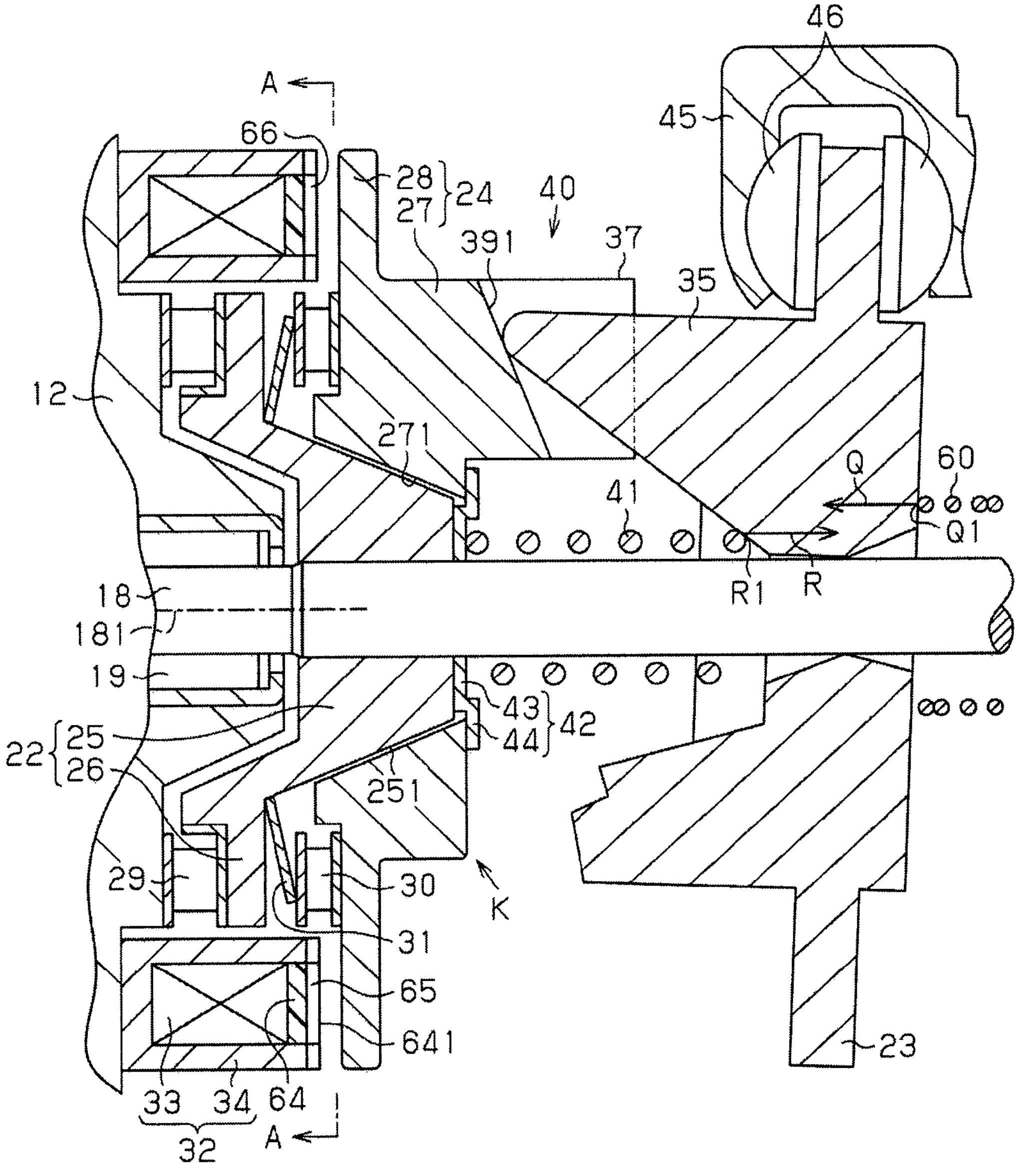
FIG. 9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A
66
28
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391
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12
271
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Q1
R1
R
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181
19
43
44
42
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25
26
251
29
30
K
31
65
641
23
33
34
64
32
A

FIG. 10

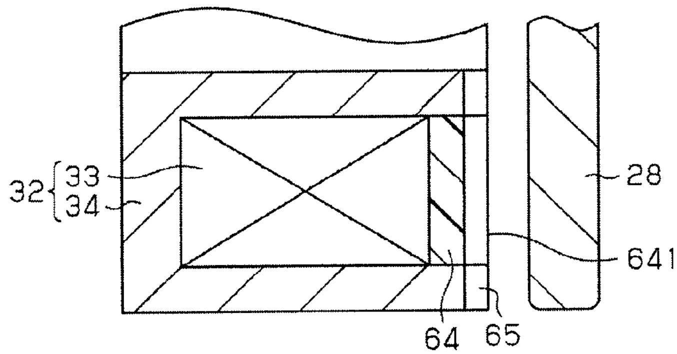
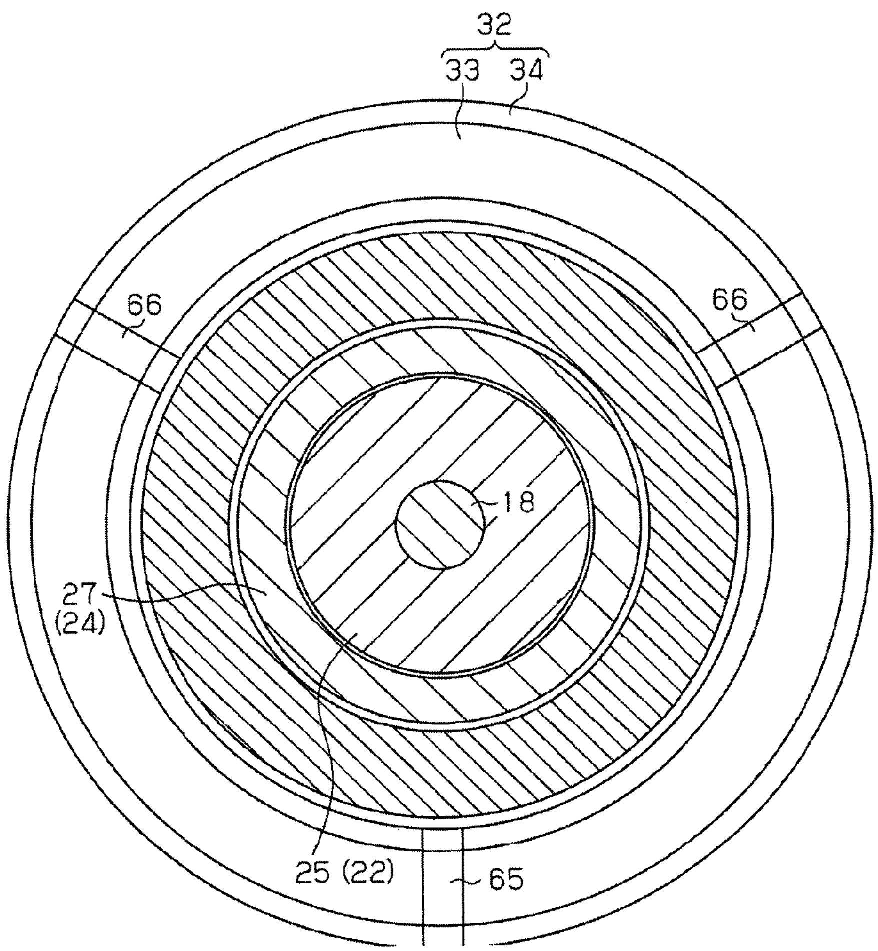



FIG. 11



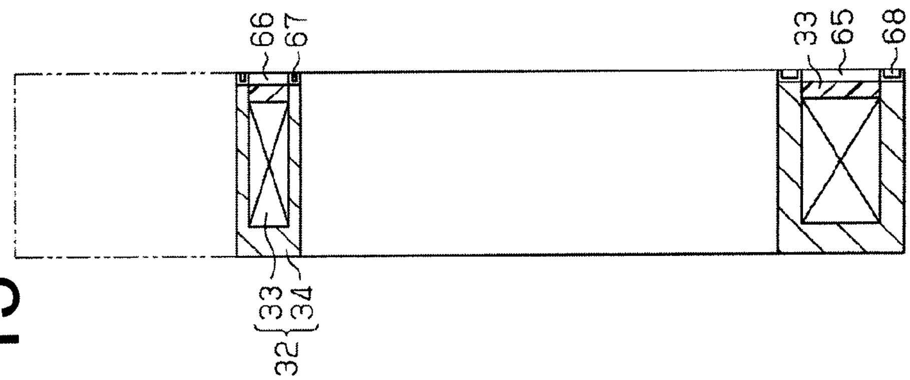



FIG. 13

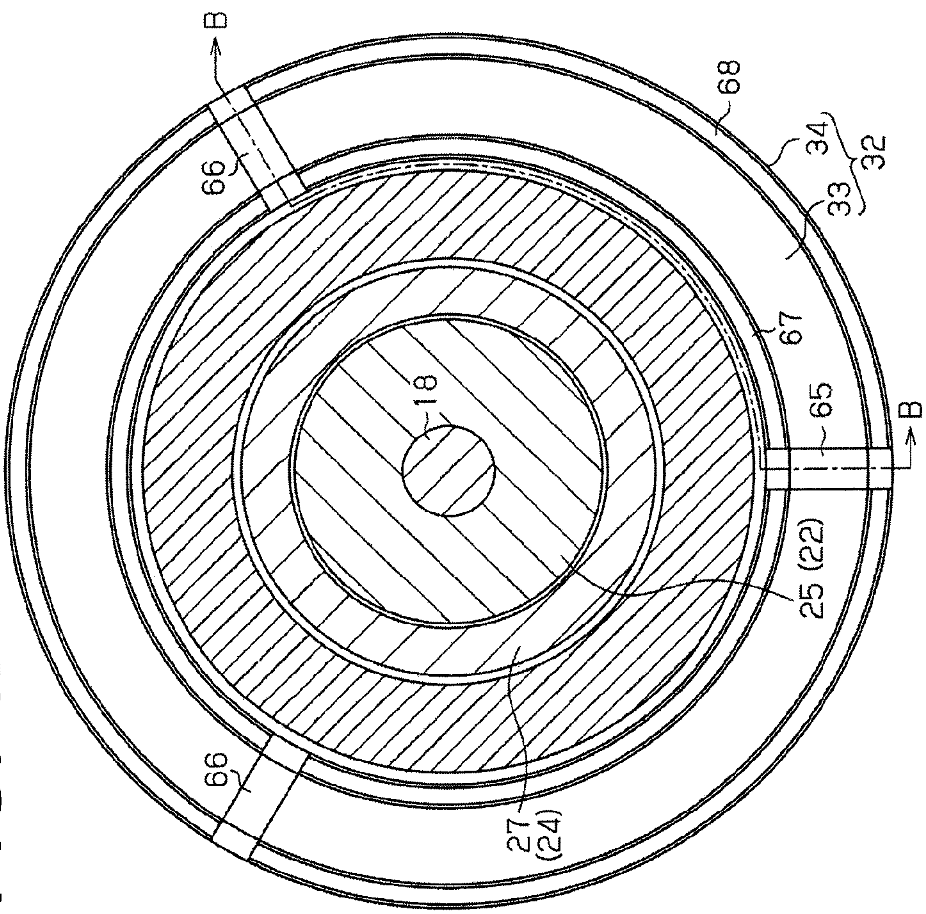
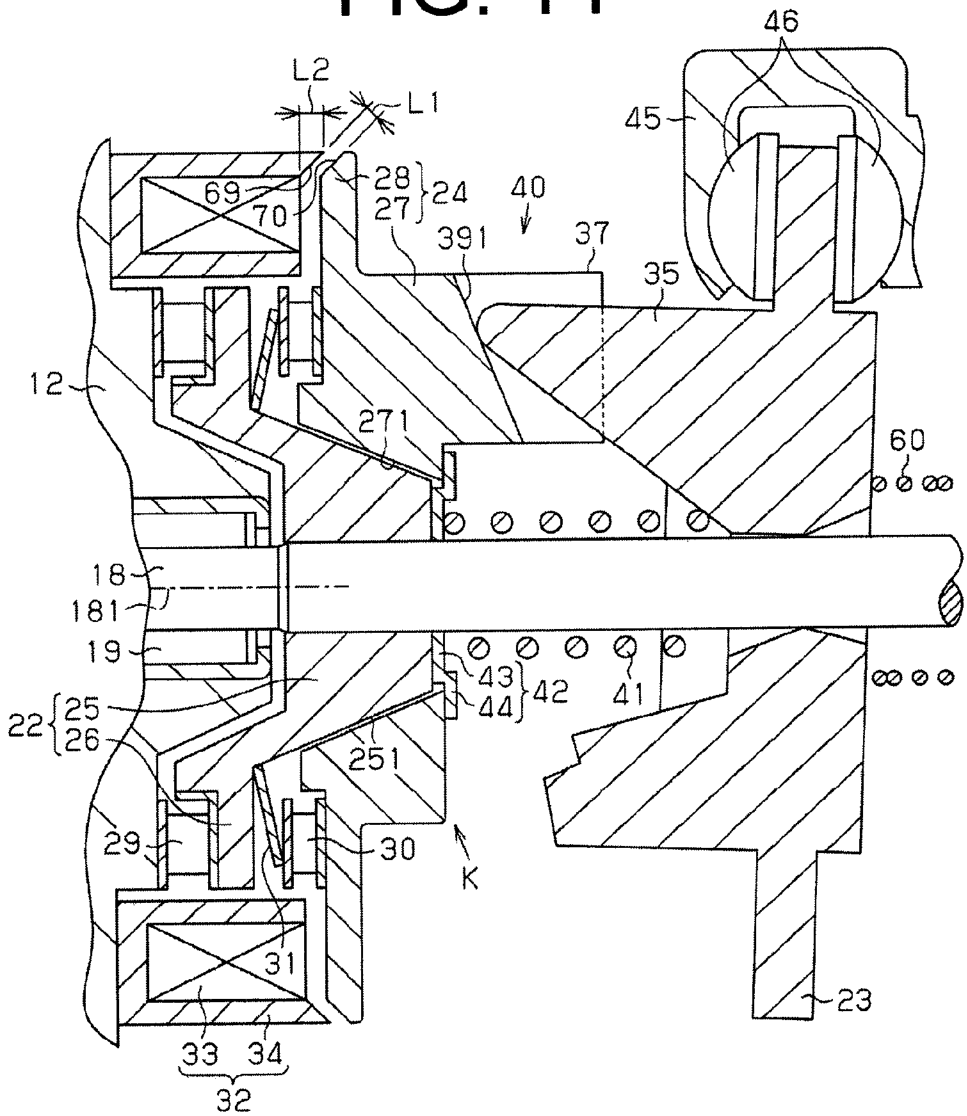



FIG. 12

FIG. 14



L2
L1
46
45
69
70
28
27
24
40
391
37
35
12
271
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42
41
22
25
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251
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K
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32
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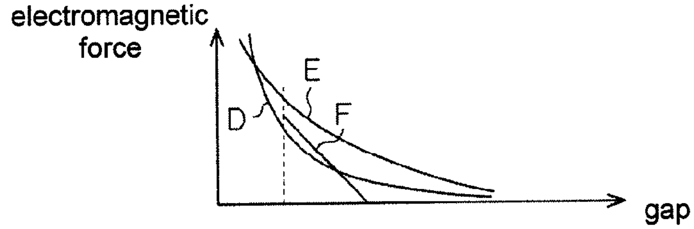
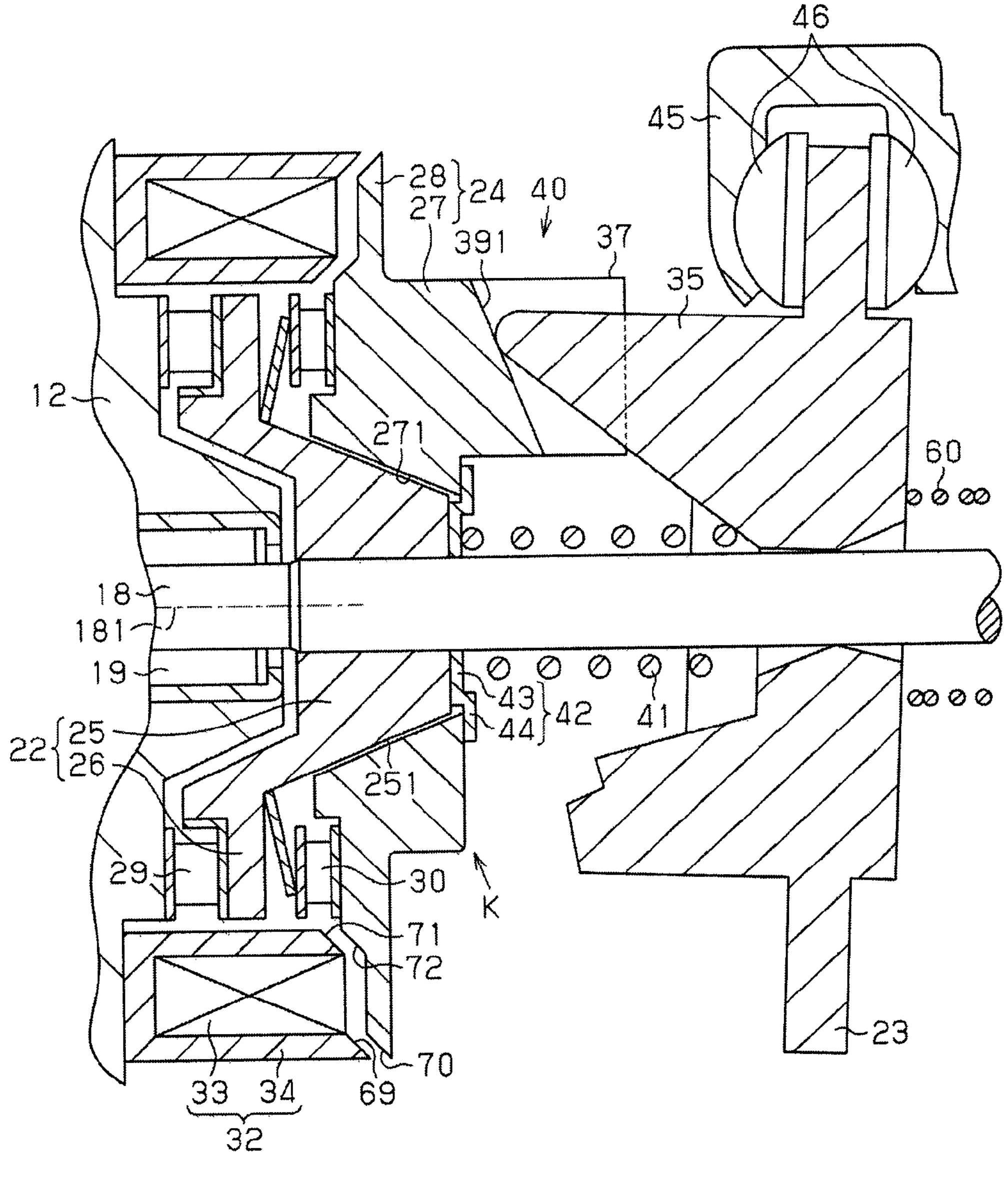
FIG. 15



electromagnetic force
E
D
F
gap

FIG. 16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24
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391
37
35
12
271
60
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181
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43
44
42
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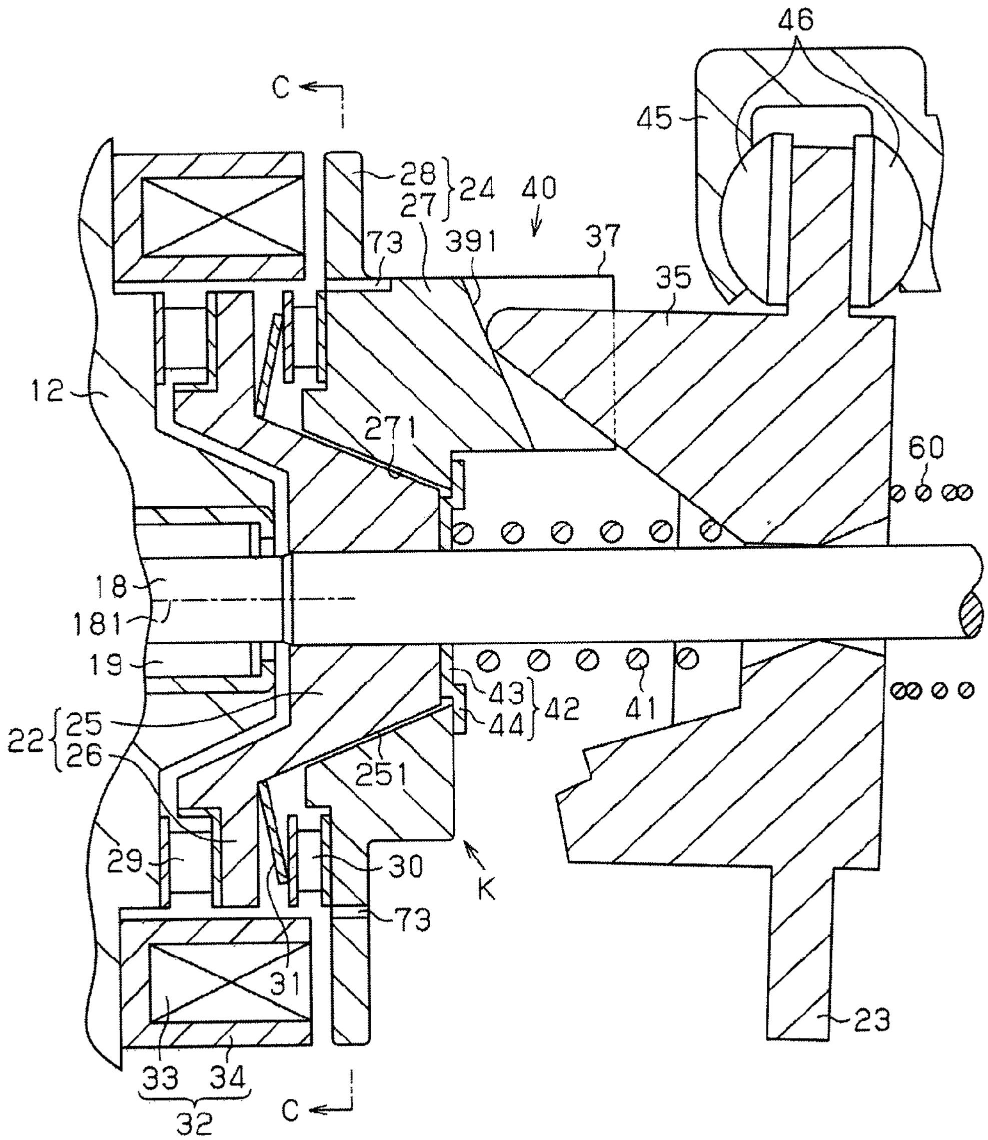
22
25
26
251
29
30
K
71
72
23
70
69
33
34
32

FIG. 17



46
45
C
28
27
24
40
73
391
37
35
12
271
60
18
181
19
43
42
41
44
22
25
26
251
29
30
K
73
31
23
33
34
32
C

FIG. 18

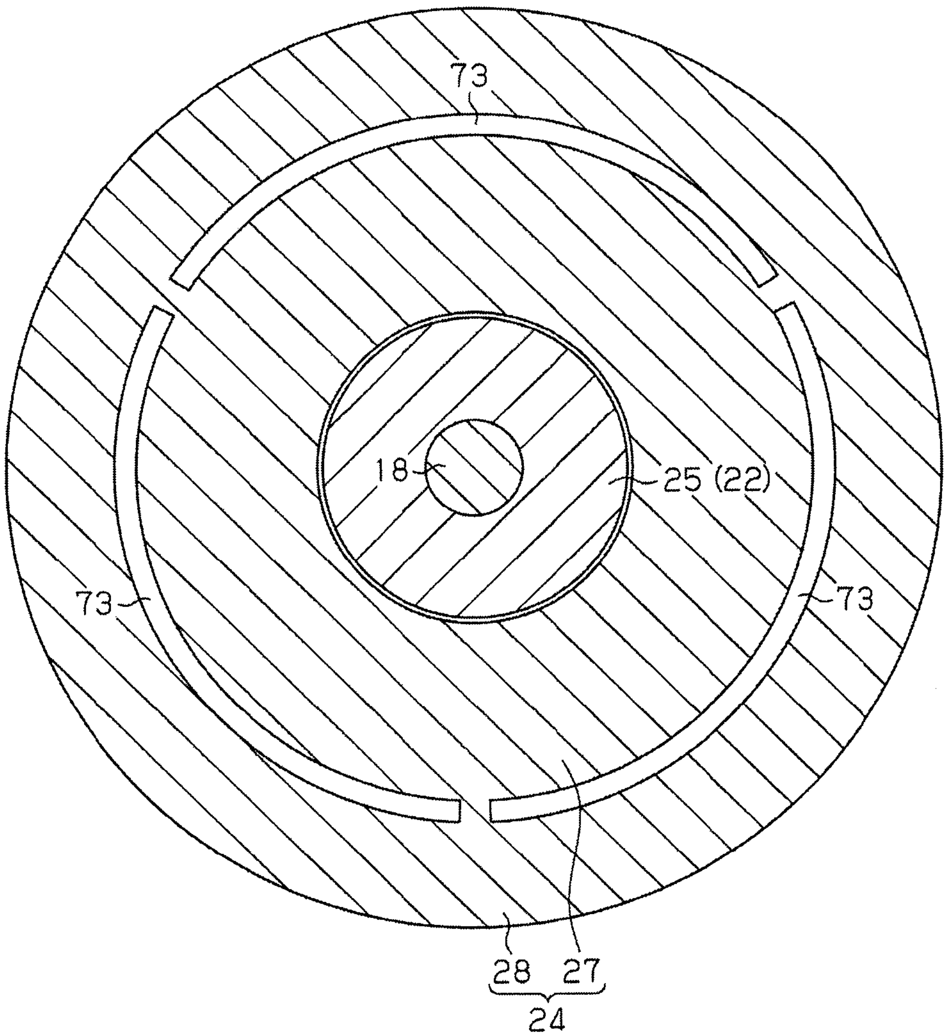
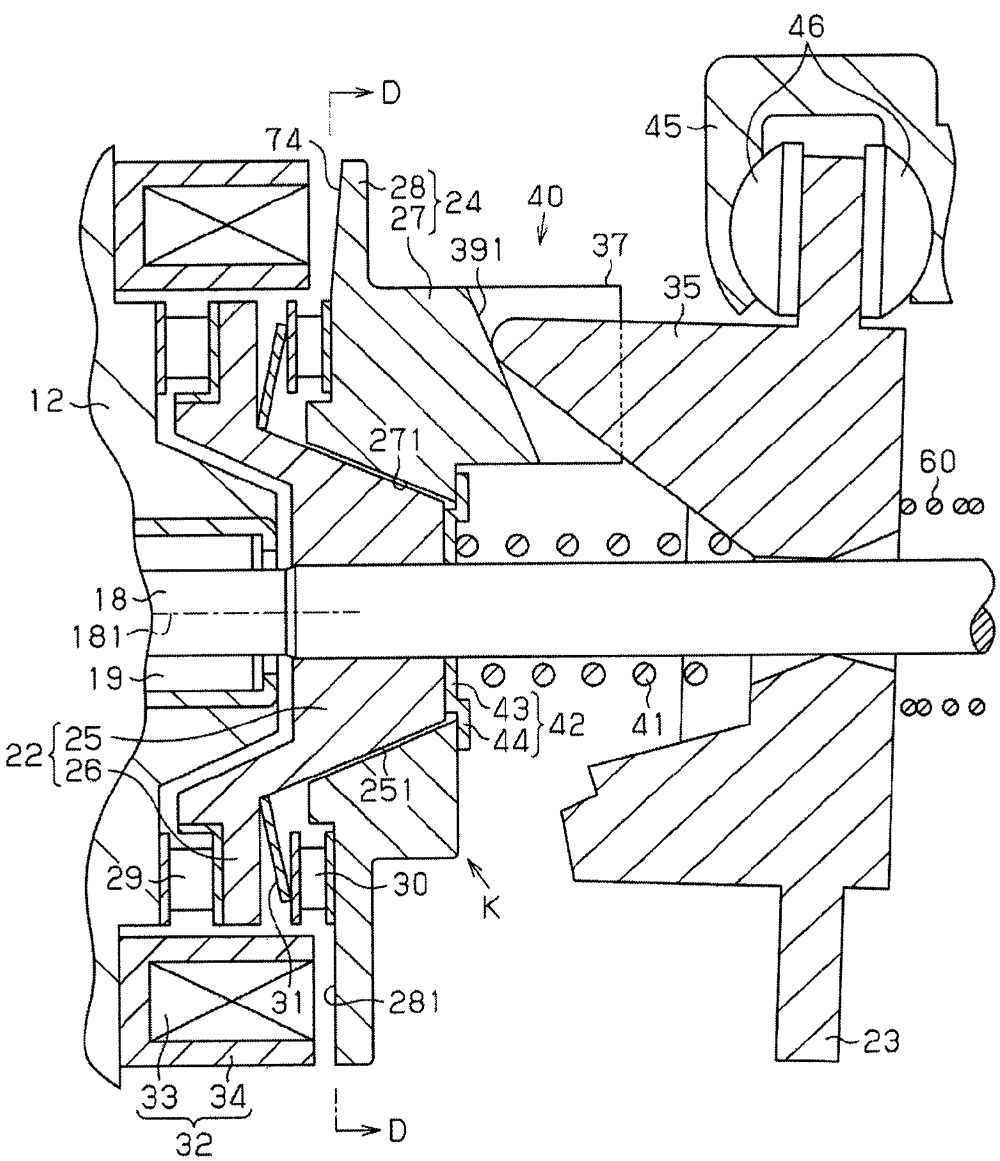


FIG. 19


46
45
D
74
28
27
24
40
391
37
35
12
271
60
18
181
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43
44
42
41
22
25
26
251
29
30
K
31
281
23
33
34
32
D

FIG. 20

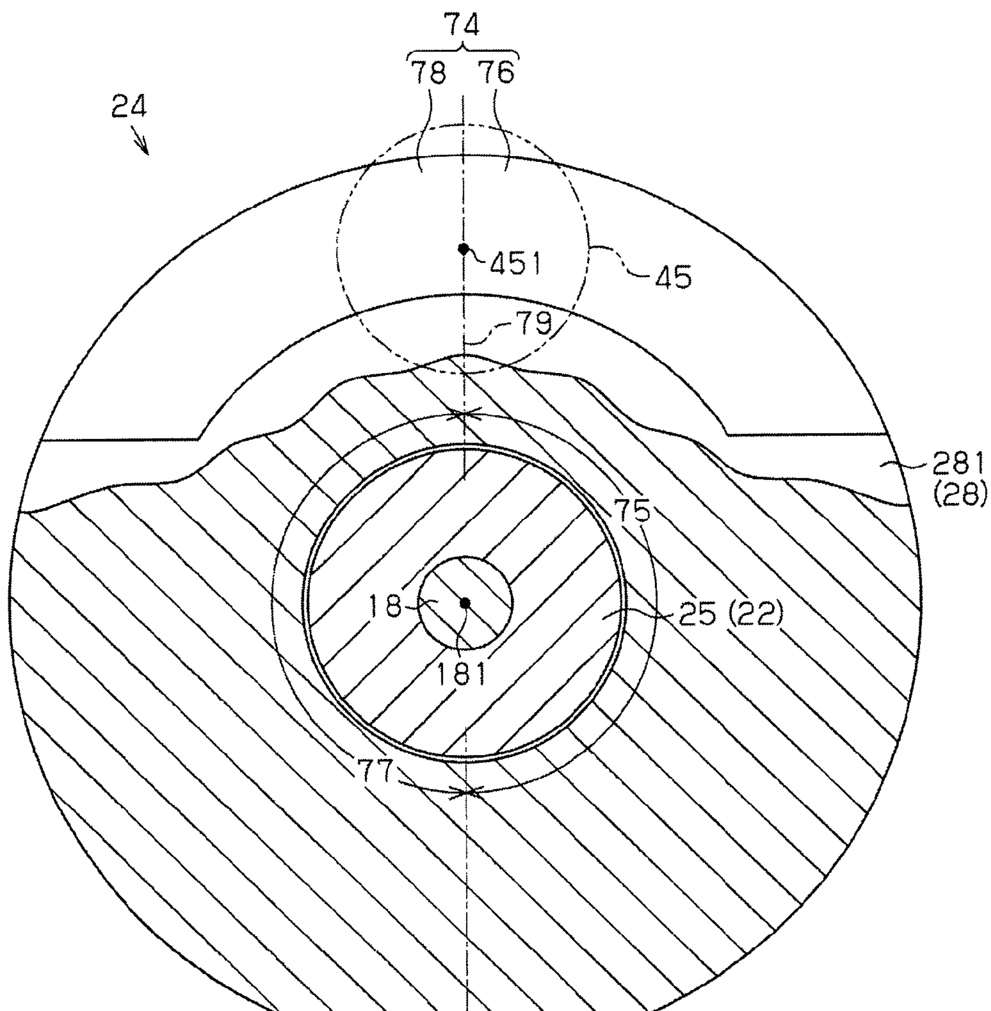


FIG. 21

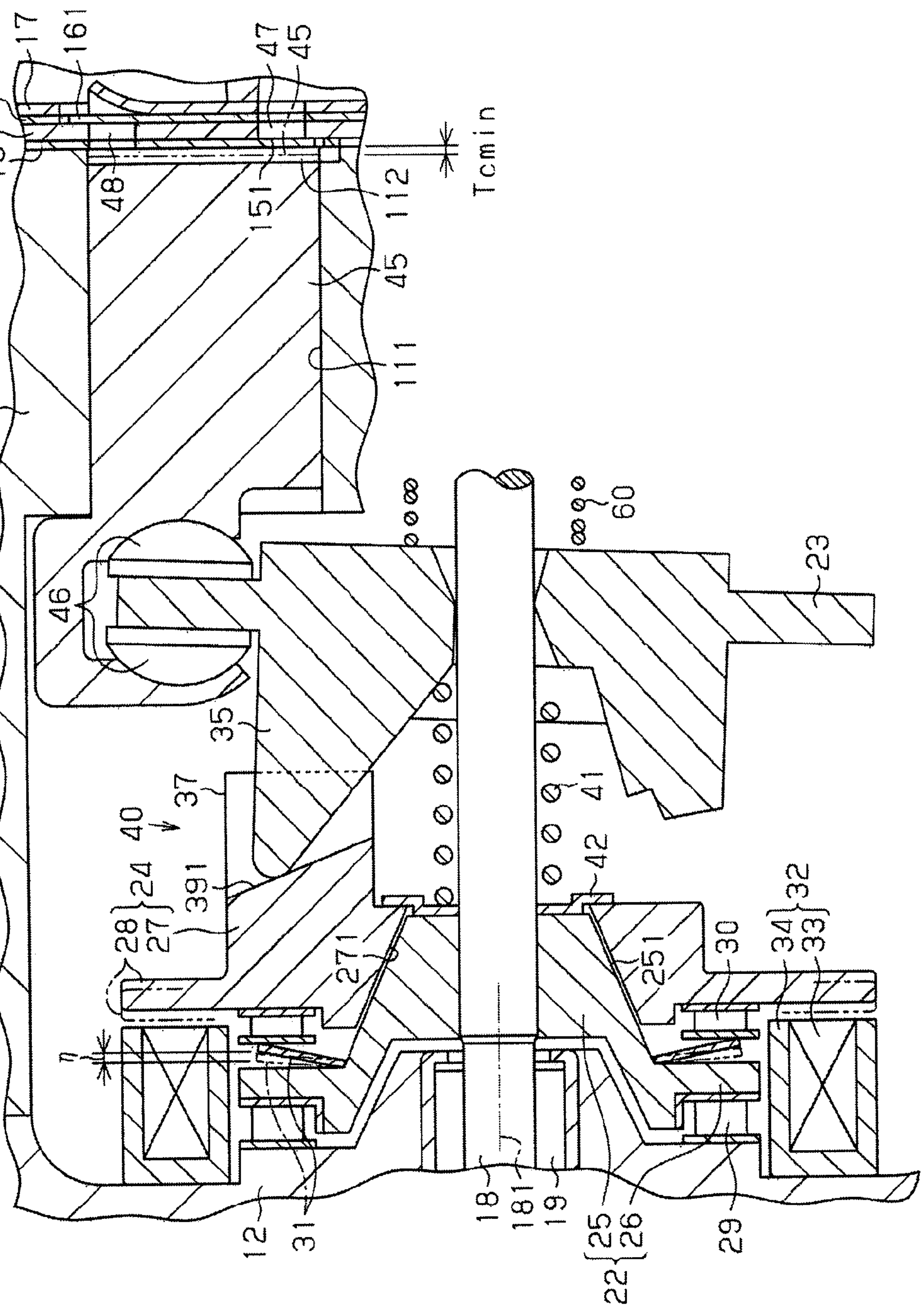

17
161
16
14
15
47
45
48
151
112
Tcmin
45
111
11
60
23
46
35
37
40
41
42
24
27
28
391
271
251
30
34
33
32
η
12
31
18
181
19
25
26
22
29

FIG. 22

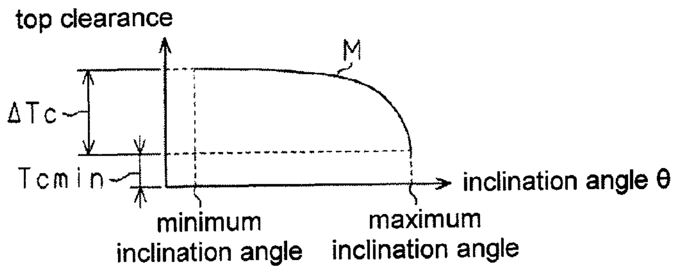
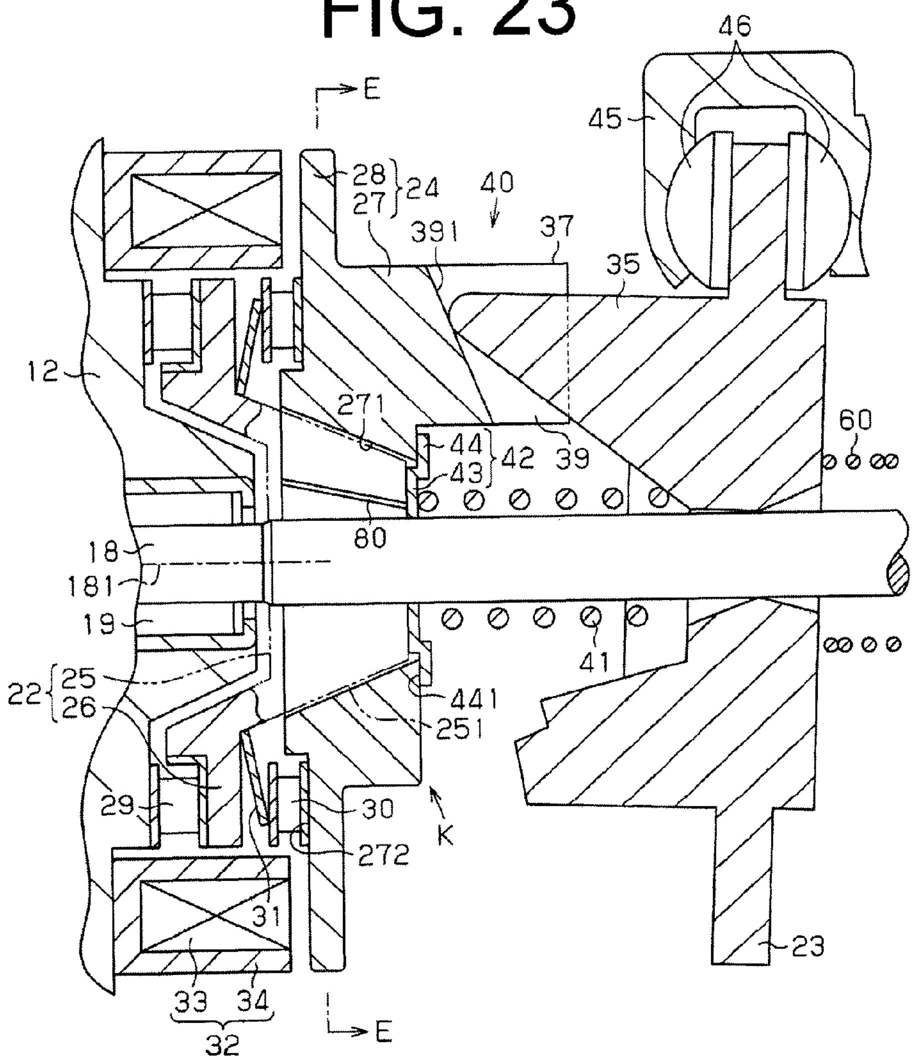

top clearance
M
ΔTc
Tcmin
inclination angle θ
minimum
inclination angle
maximum
inclination angle

FIG. 23


E
46
45
28
27
24
40
391
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35
12
271
44
43
42
39
60
18
181
80
19
22
25
26
41
441
251
29
30
272
K
31
23
33
34
32
E

FIG. 24

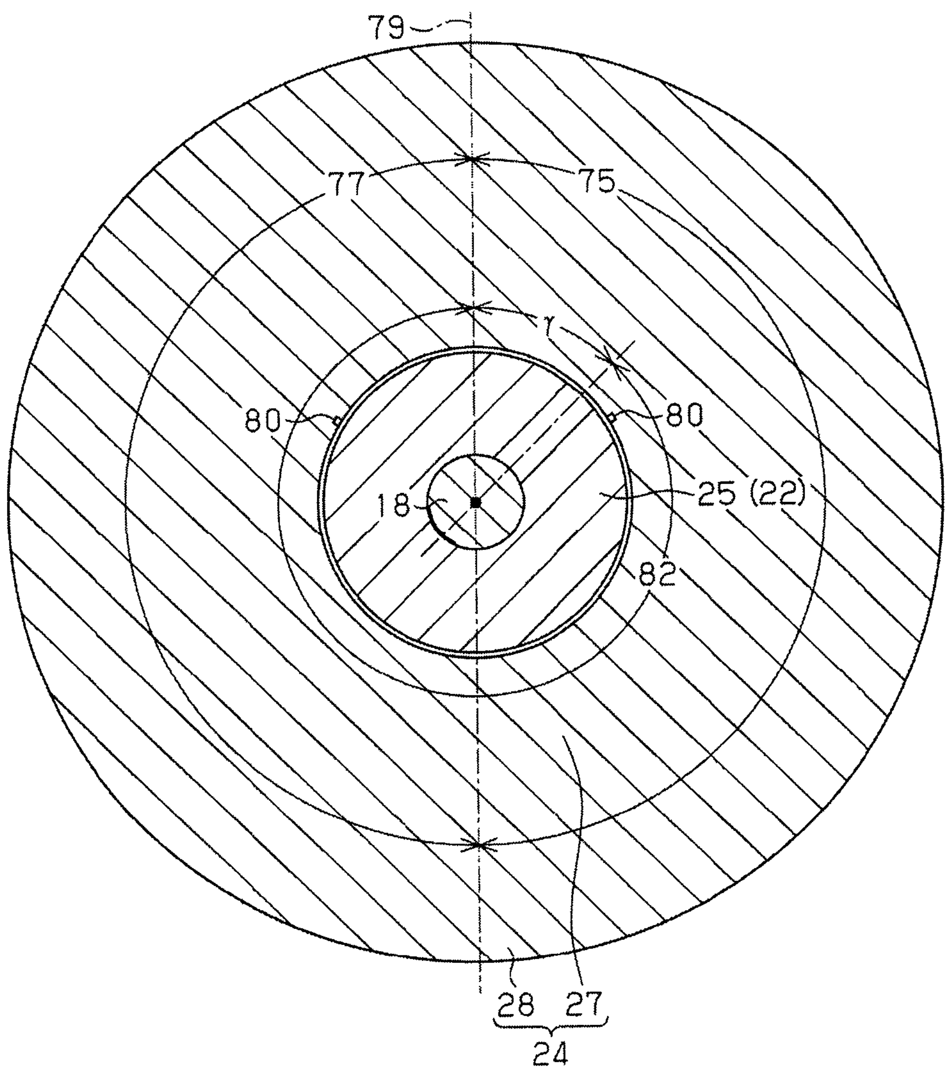


FIG. 25

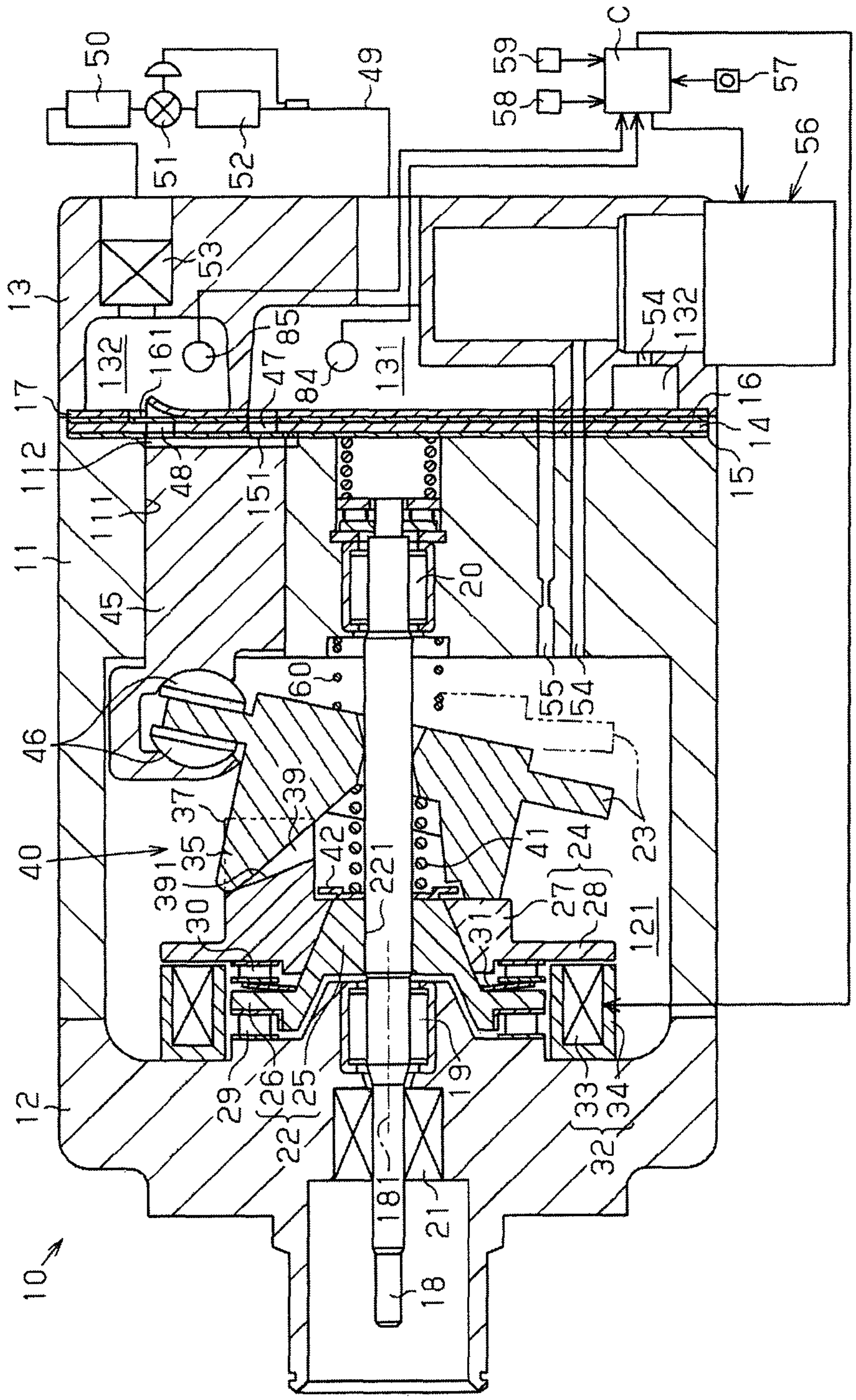


FIG. 26

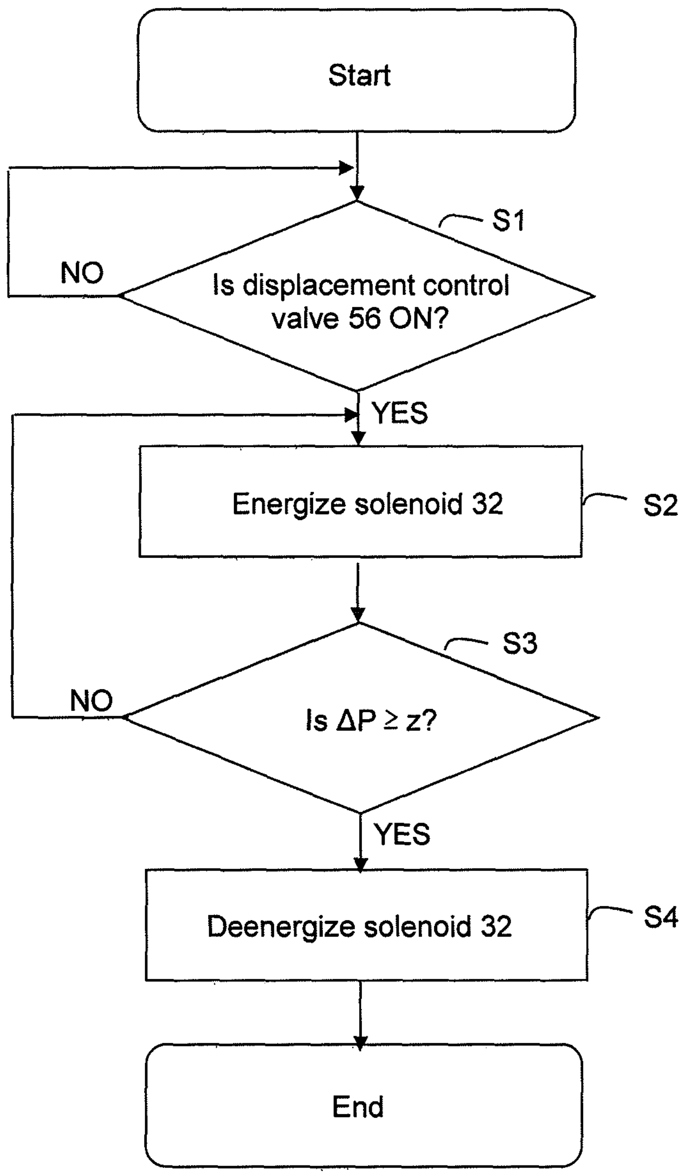
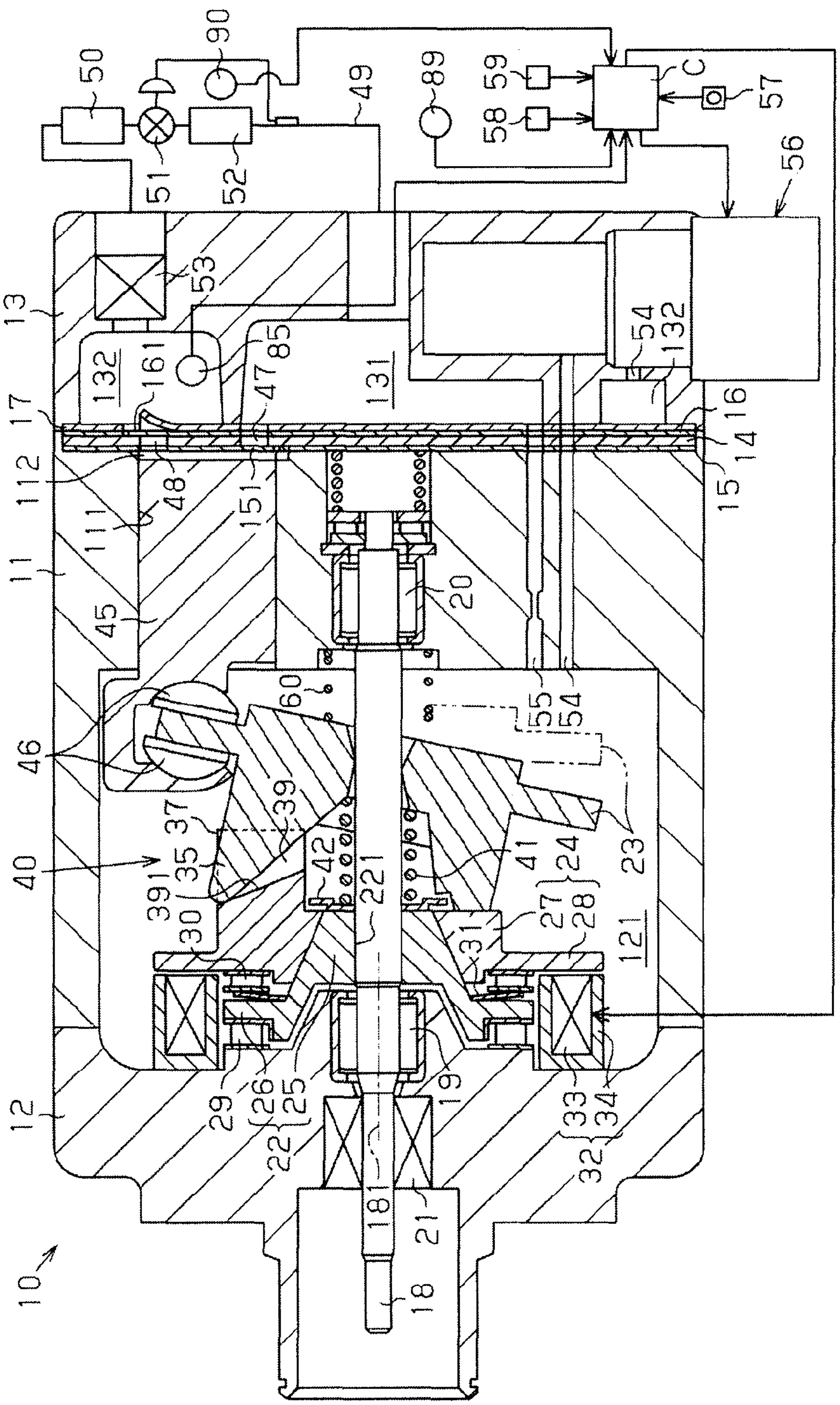

Start
S1
NO
Is displacement control valve 56 ON?
YES
Energize solenoid 32
S2
S3
NO
Is ΔP ≥ z?
YES
Deenergize solenoid 32
S4
End

FIG. 27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60
85
89
90
111
112
121
131
132
151
161
181
221
C

FIG. 28

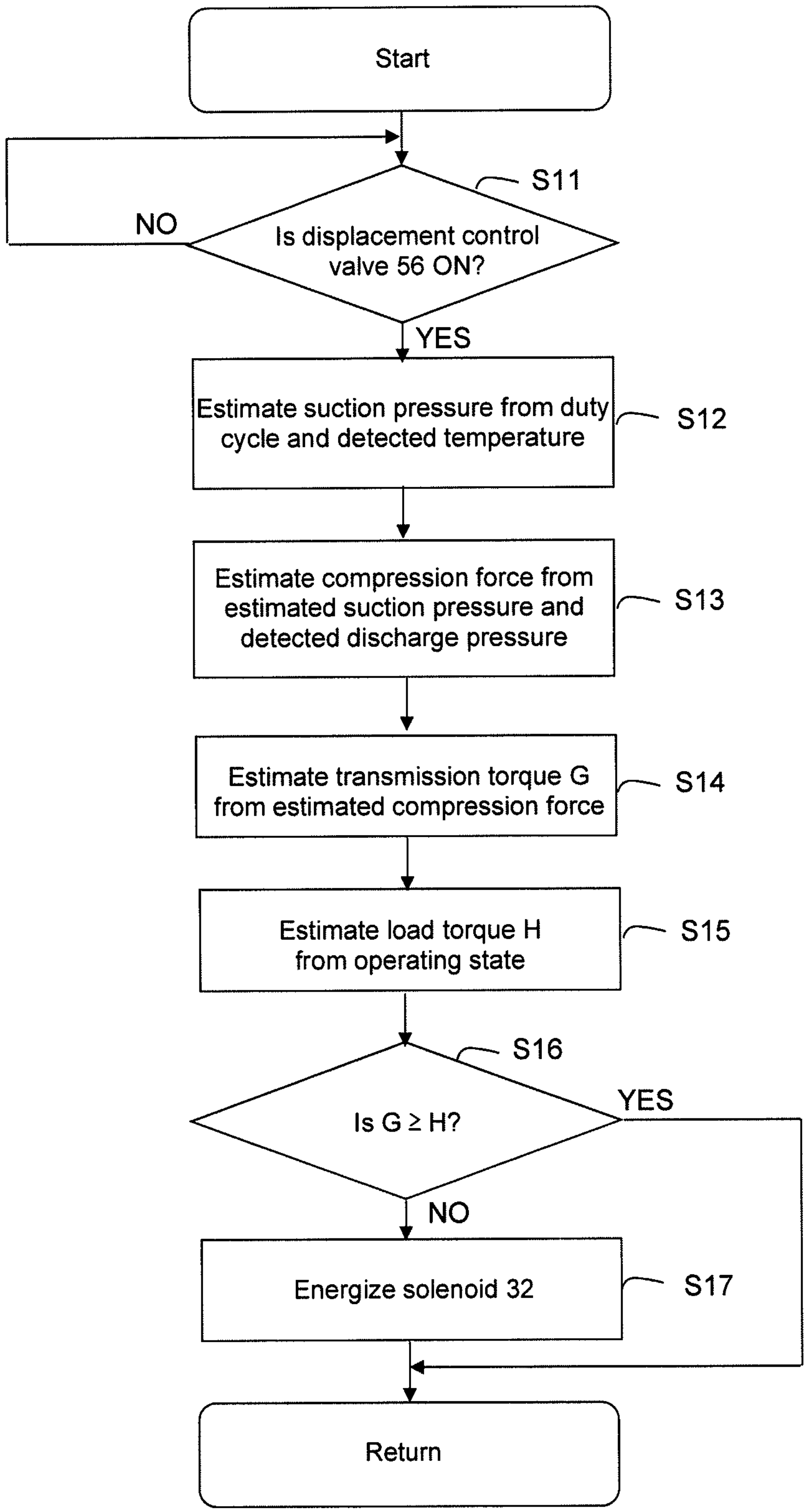
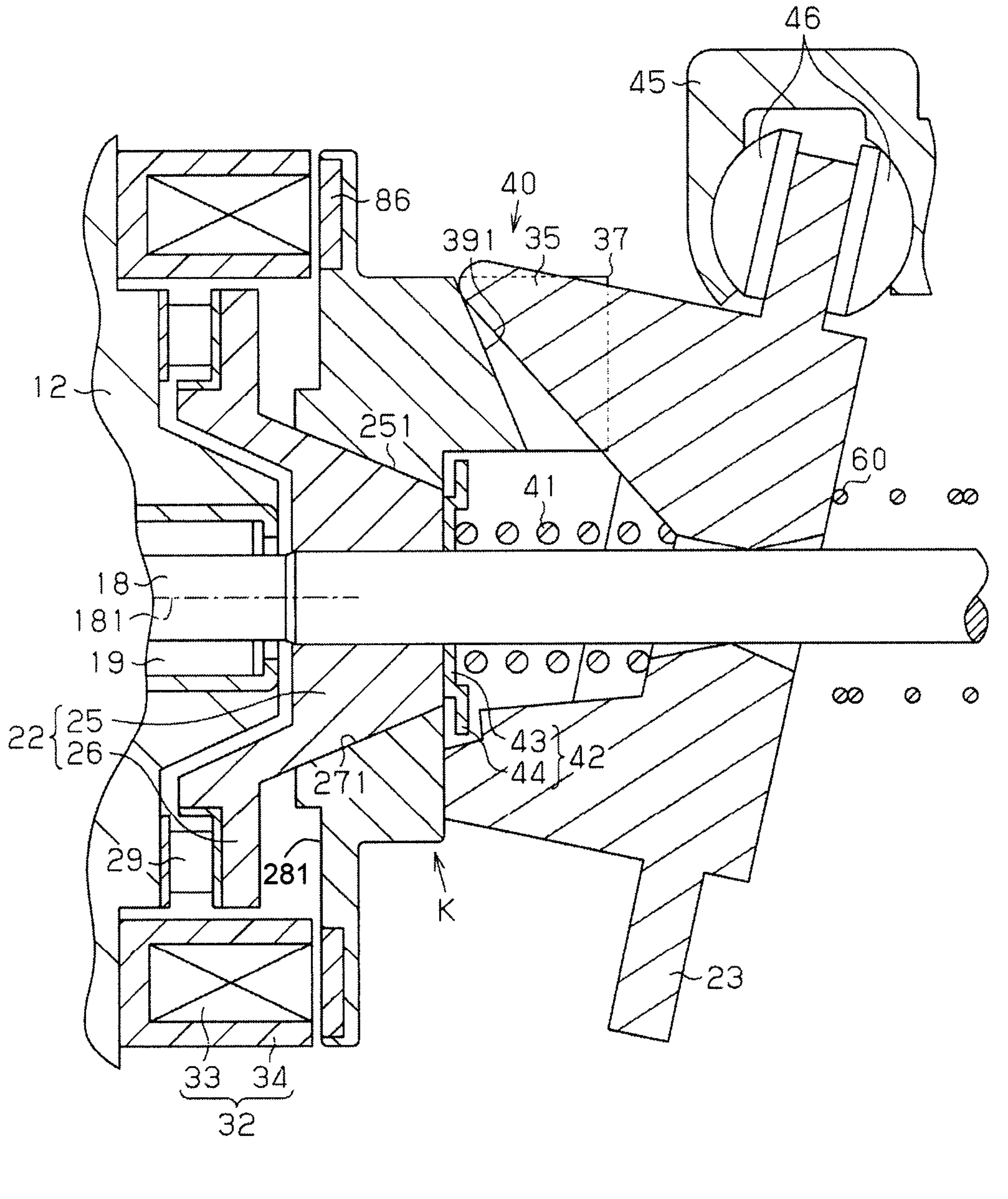

Start
S11
NO
Is displacement control valve 56 ON?
YES
Estimate suction pressure from duty cycle and detected temperature
S12
Estimate compression force from estimated suction pressure and detected discharge pressure
S13
Estimate transmission torque G from estimated compression force
S14
Estimate load torque H from operating state
S15
S16
Is G ≥ H?
YES
NO
Energize solenoid 32
S17
Return

FIG. 29


46
45
86
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391
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37
12
251
60
41
18
181
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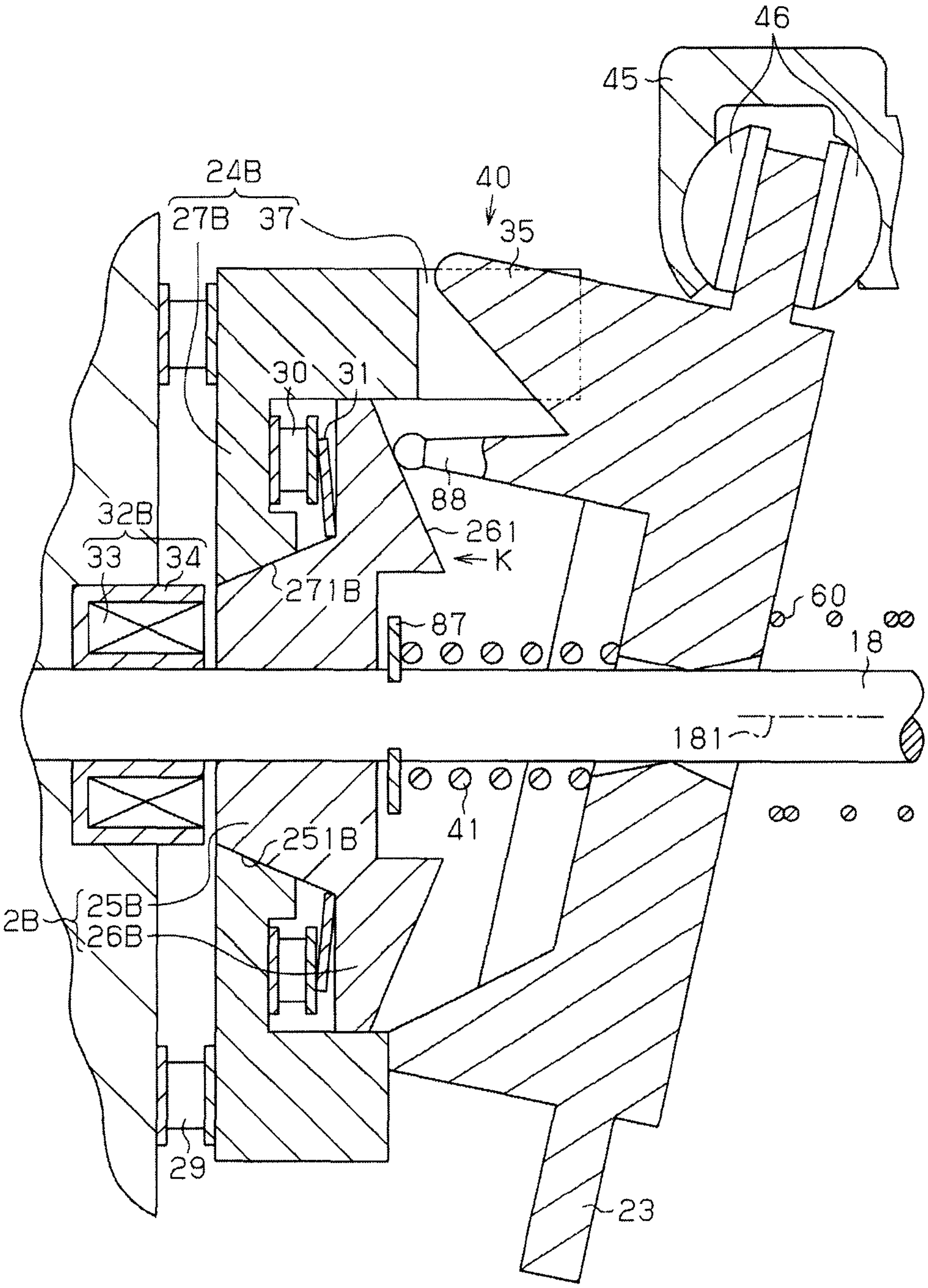
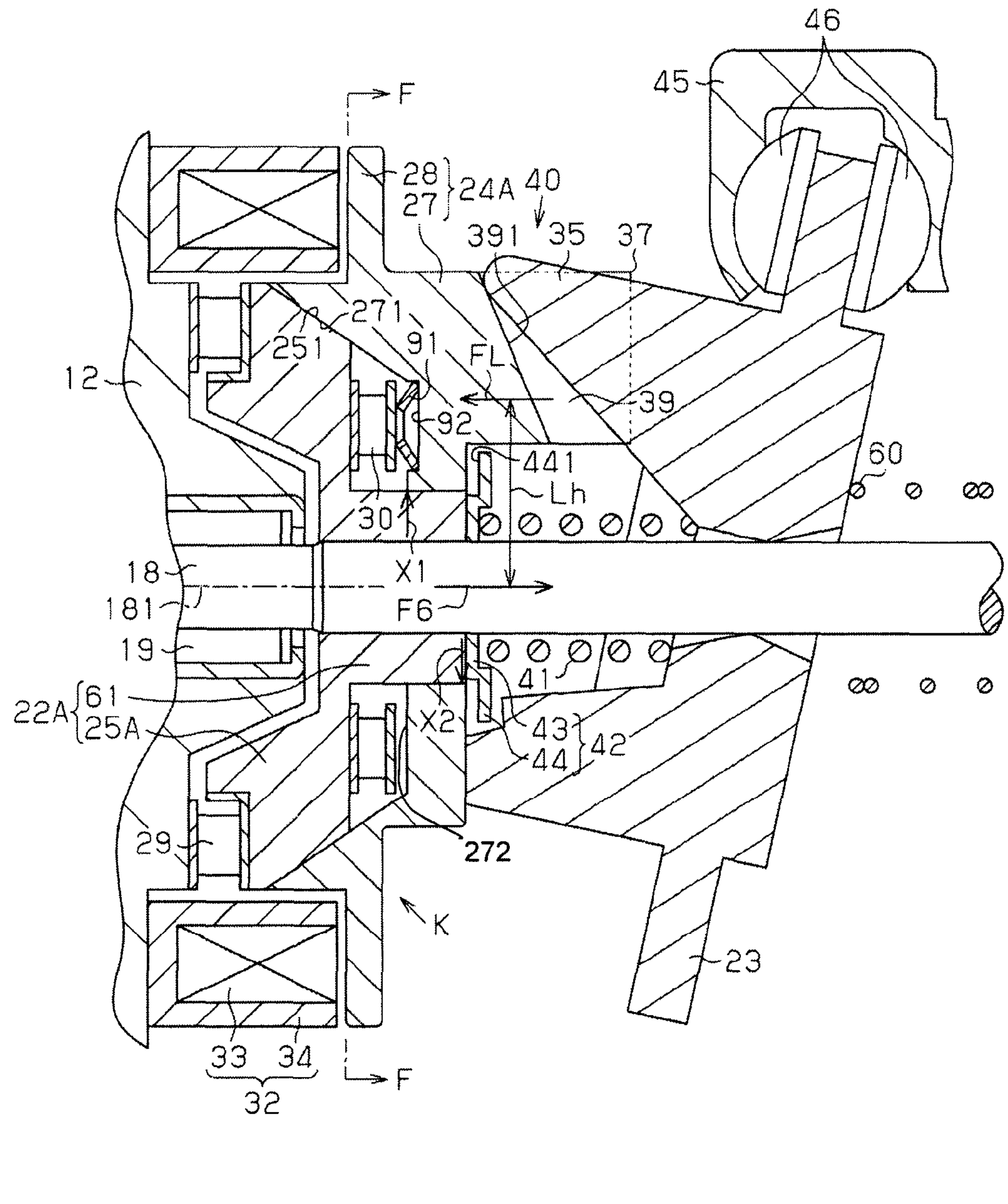
43
42
44
271
29
281
K
23
33
34
32

FIG. 30



46
45
40
24B
27B
37
35
30
31
88
261
K
32B
33
34
271B
60
87
18
181
41
251B
22B
25B
26B
29
23

FIG. 31



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45
F
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27
24A
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35
37
271
251
91
FL
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92
39
441
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Lh
30
X1
18
181
F6
19
41
22A
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25A
X2
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44
42
29
272
K
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32
F

FIG. 32

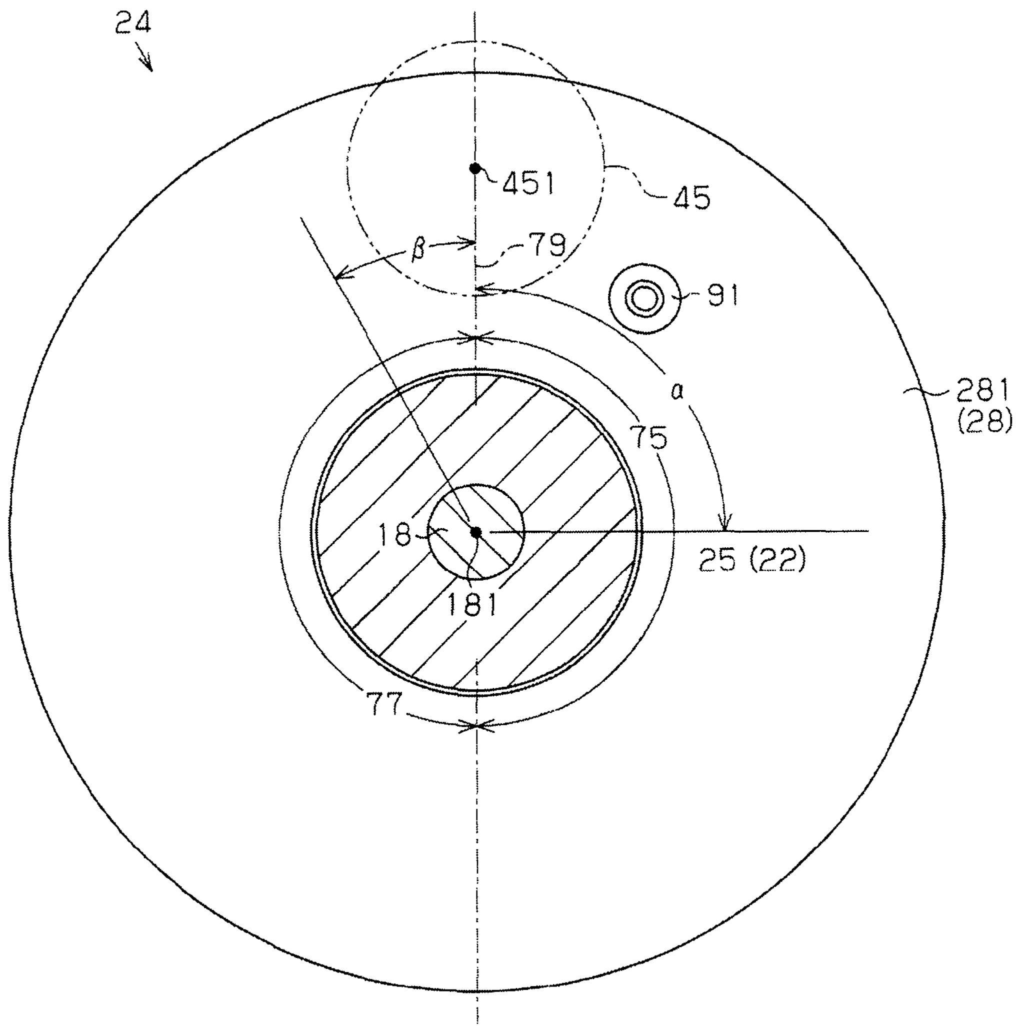
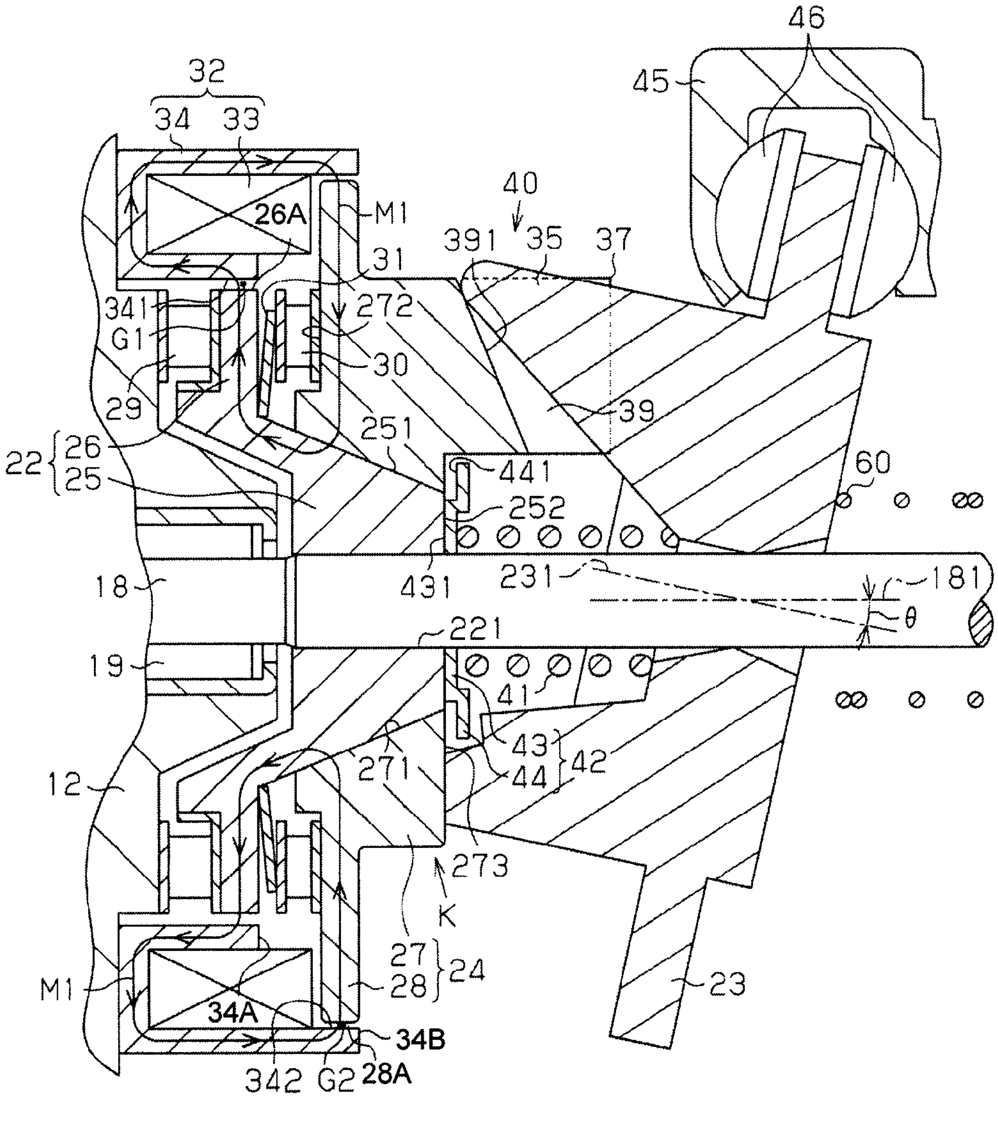


FIG. 33



32
34
33
26A
M1
40
391
35
37
31
272
341
G1
30
29
39
22
26
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251
441
252
60
18
431
231
181
θ
221
19
41
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K
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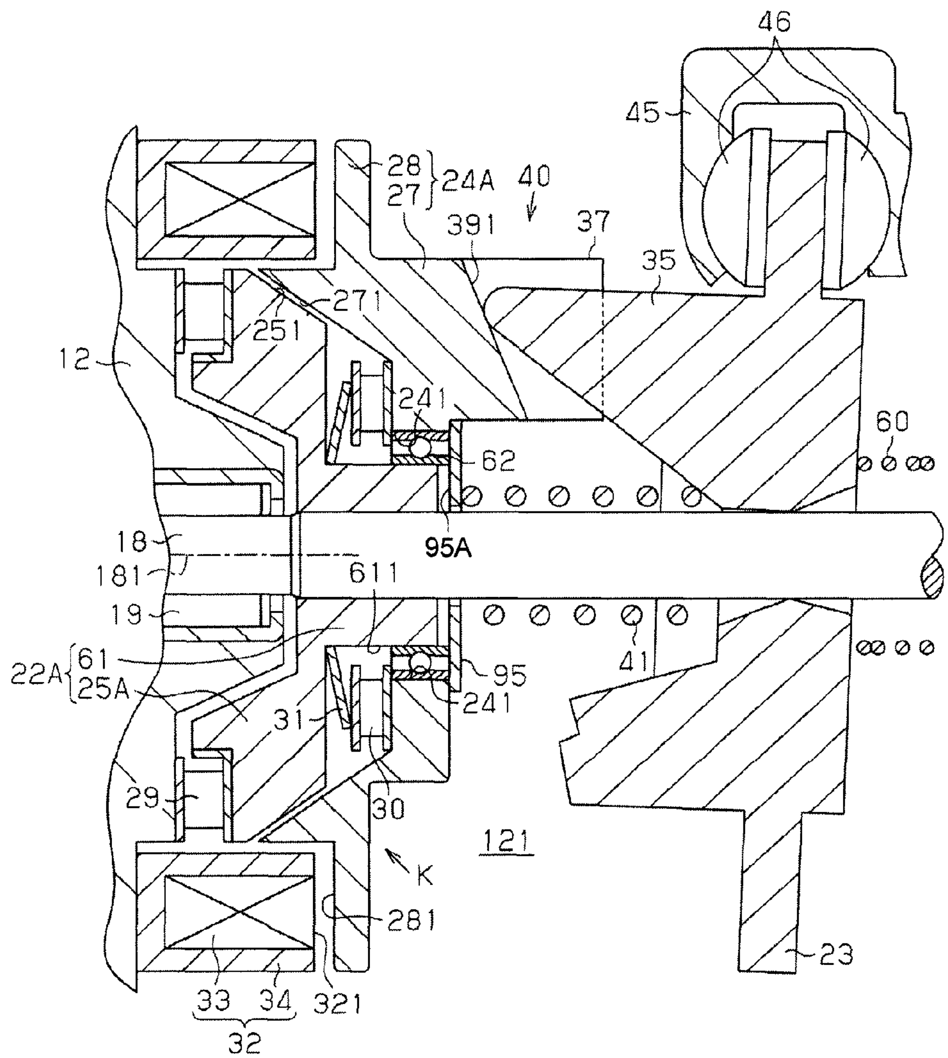
28
23
34A
34B
342
G2
28A
45
46

FIG. 34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95A
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22A
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25A
95
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31
29
30
121
K
281
23
33
34
321
32

FIG. 35

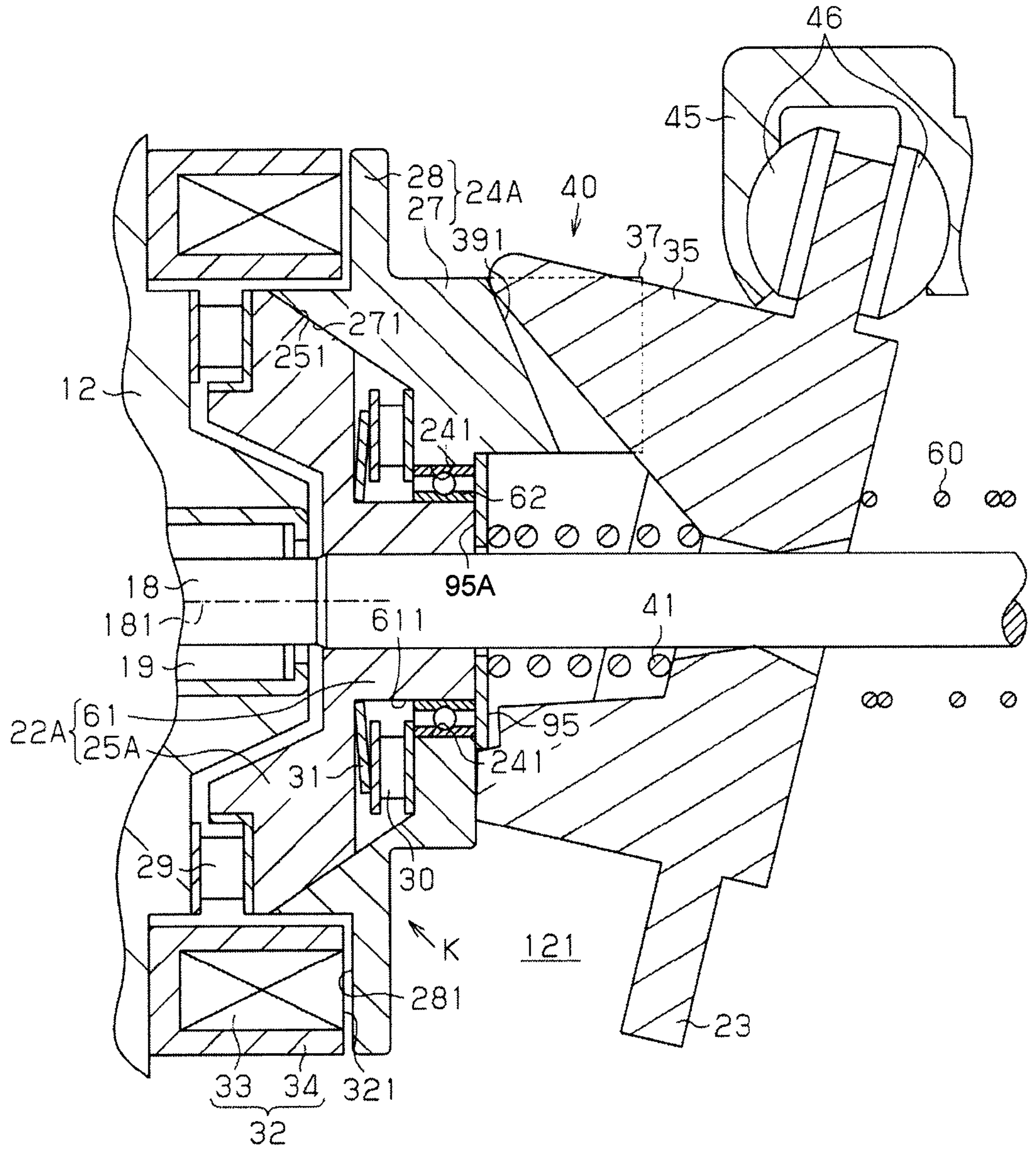
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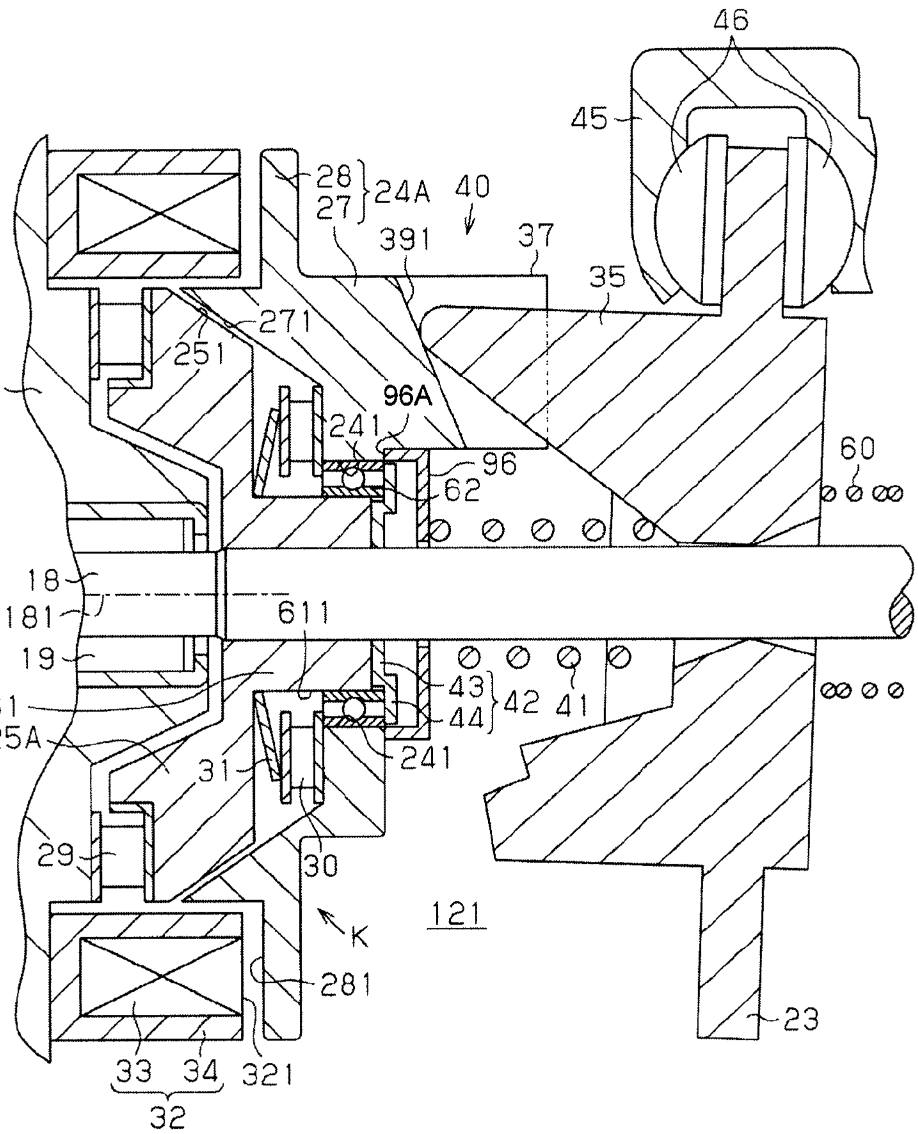
FIG. 36



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611
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22A
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25A
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FIG. 37

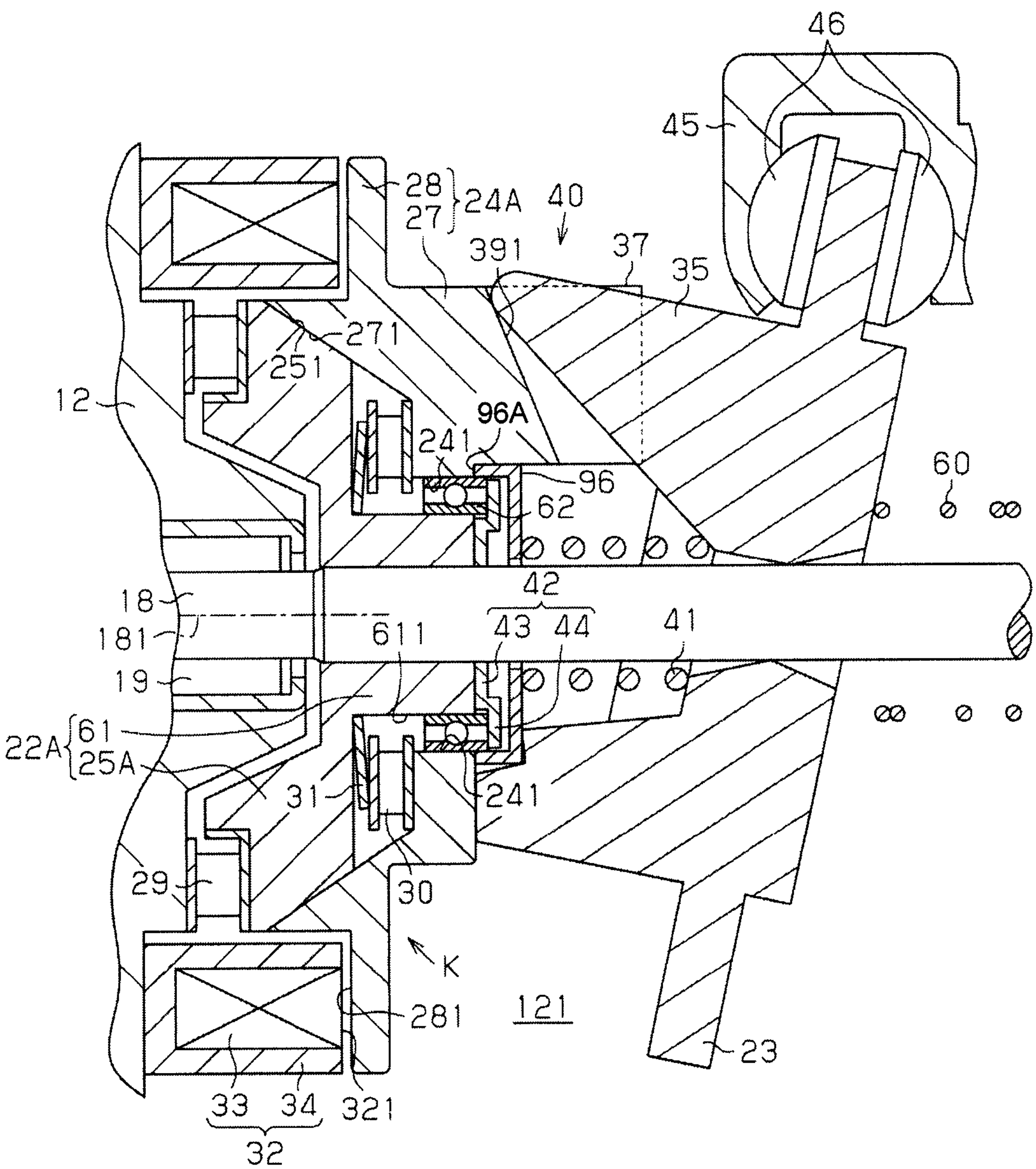
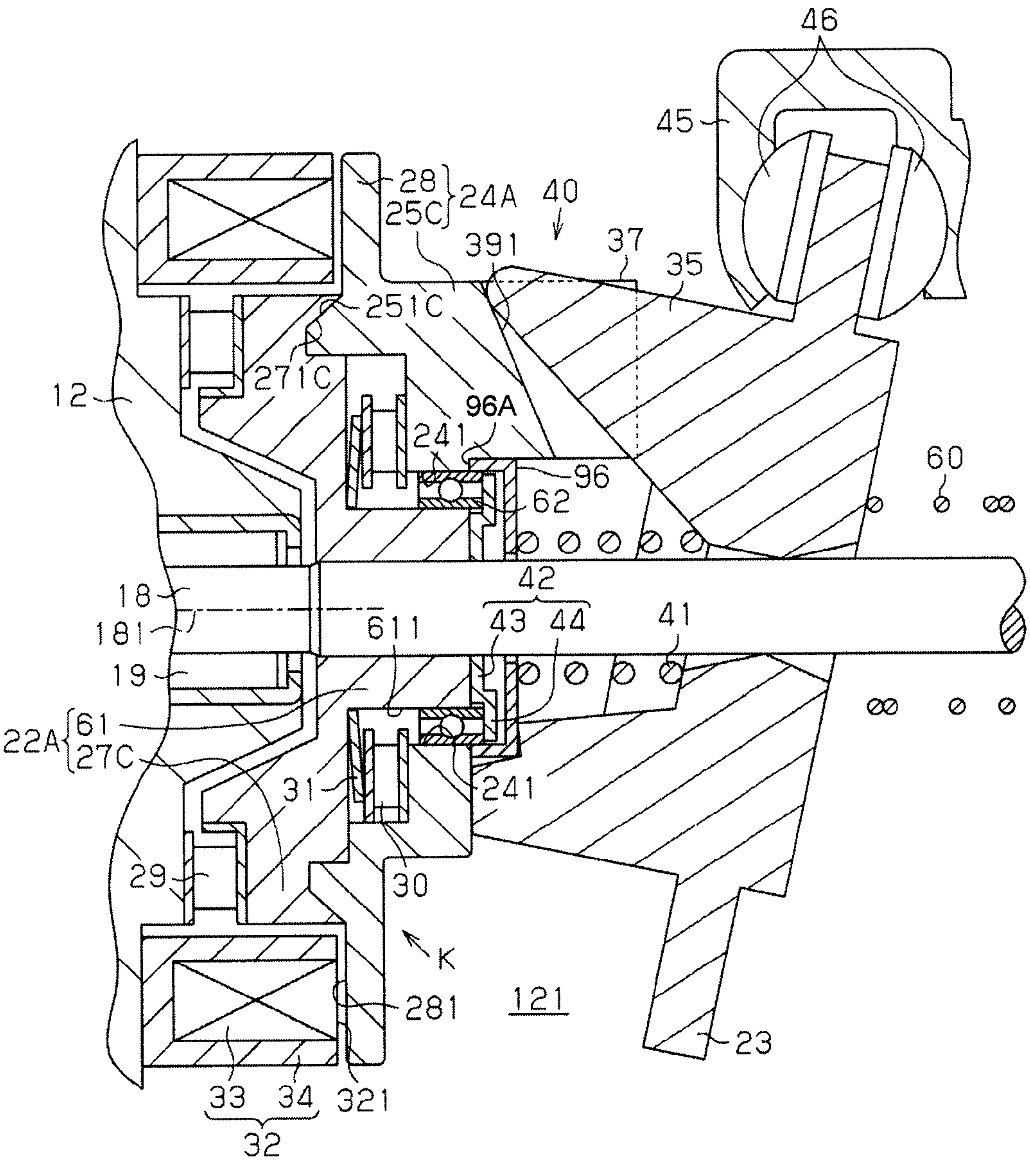

46
45
28
27
24A
40
391
37
35
271
251
12
96A
241
96
62
60
42
18
181
611
43
44
41
19
22A
61
25A
31
241
29
30
K
281
121
23
33
34
321
32

FIG. 38


46
45
28
25C
24A
40
391
37
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251C
271C
12
96A
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42
43
44
41
611
19
22A
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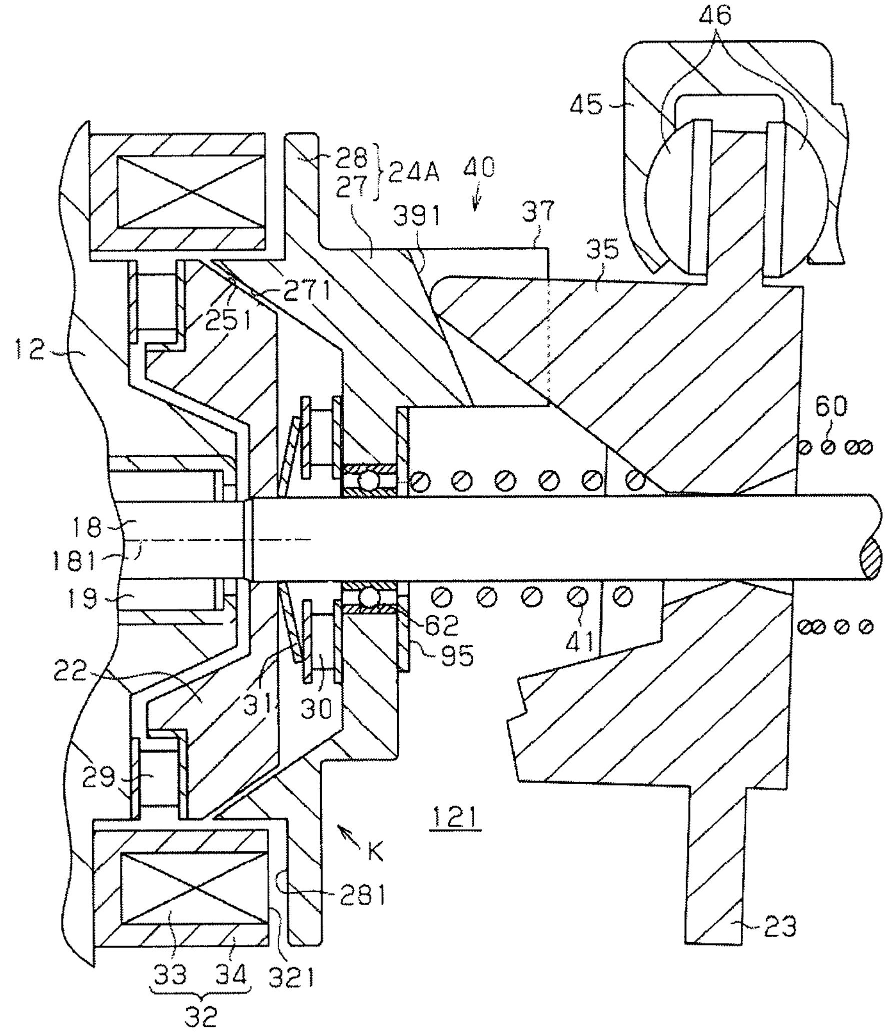
27C
31
241
29
30
K
281
121
23
33
34
321
32

FIG. 39


46
45
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24A
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271
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181
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41
22
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# SWASH PLATE TYPE VARIABLE DISPLACEMENT COMPRESSOR AND METHOD OF CONTROLLING SOLENOID THEREOF

## BACKGROUND OF THE INVENTION

The present invention relates to a swash plate type variable displacement compressor including a rotary shaft, a swash plate and a plurality of pistons, wherein the swash plate is rotated by driving force of the rotary shaft, the swash plate being inclinable at a variable inclination angle, the pistons being engaged with the swash plate and reciprocable in accordance with the rotation of the swash plate so that the length of the stroke of each piston is varied depending on the inclination angle of the swash plate. The present invention also relates to a method of controlling a solenoid of the swash plate type variable displacement compressor.

In the swash plate type variable displacement compressor, when the swash plate is rotated by the rotation of the rotary shaft, the rotation of the swash plate is transmitted to the pistons via pairs of shoes thereby to cause the reciprocating motion of the pistons for compressing a refrigerant. When the inclination angle of the swash plate is changed with respect to the rotary shaft, the length of the stroke of each piston is changed thereby to vary the displacement of the swash plate type variable displacement compressor.

The swash plate type variable displacement compressor which is disclosed by Japanese Unexamined Patent Application Publication No. 2007-24257 has an electromagnetic clutch as the power transmission mechanism between the rotary shaft and the engine. The electromagnetic clutch is provided outside the compressor housing of the swash plate type variable displacement compressor. If the electromagnetic clutch is not used as the power transmission mechanism, power produced by the engine may be transmitted to the rotary shaft at all times. In such a swash plate type variable displacement compressor having no electromagnetic clutch, the engine rotates the rotary shaft constantly. In a vehicle air conditioner, therefore, when cooling operation is not needed, the displacement of the swash plate type variable displacement compressor is minimized by keeping the swash plate at the minimum inclination angle position. The minimization of the displacement reduces the load applied to the engine thereby to improve the fuel efficiency of the engine.

In the swash plate type variable displacement compressor with or without the electromagnetic clutch has pairs of shoes which are disposed in sliding contact with the swash plate. The sliding resistance between the shoes and the swash plate causes a mechanical loss, thereby providing the additional load applied to the engine. Particularly in a swash plate type variable displacement compressor having no clutch, the mechanical loss caused by the sliding resistance needs to be reduced in order to reduce the load applied to the engine when the compressor is operated at its minimum displacement (or with the swash plate placed at the minimum inclination angle position).

In the swash plate type variable displacement compressor which is disclosed by Japanese Unexamined Patent Application Publication No. 2006-152918, the swash plate is supported by its support that rotates integrally with the rotary shaft. The swash plate and the support are connectable to and disconnectable from each other via a clutch. The clutch is operable between a first state (or the engaged state) where the swash plate and the support are rotated integrally and a second state (or the disengaged state) where the swash plate is rotatable relative to the support. The spring force of a compression spring provided in the support and the centrifugal force acting on a spherical body provided between the swash plate and the support urge the swash plate in the direction that causes the power transmitting portion of the support and the power receiving portion of the swash plate to be disengaged from each other. By so constructing the compressor, shifting may be done between the first state of the clutch where the priority is given to the improvement of the displacement controllability when the swash plate is placed other than at the minimum inclination angle position, and the second state of the clutch where the priority is given to the reduction of the rotational resistance when the swash plate is placed at the minimum inclination angle position.

If the power transmitting portion of the support and the power receiving portion of the swash plate are disengaged from each other when the swash plate is placed at the minimum inclination angle position, the above-described problem raised by the swash plate type variable displacement compressor having no clutch is resolved. In the case of the swash plate type variable displacement compressor having the electromagnetic clutch, the disadvantage of a large power consumption due to energization of the electromagnetic clutch is also avoided.

However, the load that urges the swash plate toward the support when the swash plate is placed at the minimum inclination angle position depends on the rotational speed of the rotary shaft. The urging load is reduced and then increased with an increase of the rotational speed of the rotary shaft. For allowing the clutch to be shifted from the second state (or the disengaged state) where the swash plate and the support are disconnected from each other to the first state (or the engaged state) where the swash plate and the support are connected to each other, therefore, the spring load of the compression spring needs to be reduced approximately to the minimum value of the urging load. In the case of a relatively low rotational speed of the rotary shaft or a rotational speed of the compressor during the idling of a vehicle where the centrifugal force acting on the spherical body is relatively small, such a spring load of the compression spring cannot release the clutch from the engaged state, so that the mechanical loss incurred while the rotary shaft is rotating at a relatively low speed, e.g. idling operation of the engine, may not be reduced.

The present invention is directed to providing a swash plate type variable displacement compressor having an electromagnetic clutch that reduces the mechanical loss and the power consumption.

## SUMMARY OF THE INVENTION

In accordance with a first aspect of the present invention, there is provided a swash plate type variable displacement compressor that includes a rotary shaft, a swash plate, a plurality of pistons, a first rotor, a second rotor, a solenoid and a cone clutch. The swash plate is rotated by driving force of the rotary shaft. The swash plate is inclinable at a variable inclination angle. The pistons are engaged with the swash plate and reciprocable in accordance with the rotation of the swash plate so that a length of stroke of each piston is varied depending on the inclination angle of the swash plate. The first rotor is connected to the rotary shaft for rotation therewith. The second rotor transmits the rotation of the first rotor to the swash plate. The solenoid produces electromagnetic force that acts on the first rotor or the second rotor so that the first rotor and the second rotor move toward each other. The cone clutch is engageable by energization of the solenoid. The cone clutch has a male cone portion and a female cone portion. The male cone portion has a conical surface provided on one of the first rotor and the second rotor. The female cone portion has a conical surface provided on the other of the first rotor and the second rotor. The conical surface of the female cone portion is connectable to and disconnectable from the conical surface of the male cone portion.

In accordance with a second aspect of the present invention, there is provided a method of controlling a solenoid of a swash plate type variable displacement compressor having a cone clutch engageable by energization of the solenoid. The method includes the steps of starting passing an electric current through the solenoid, detecting a differential pressure between a discharge pressure and a suction pressure after the step of starting passing the electric current through the solenoid, and stopping passing the electric current through the solenoid if the differential pressure reaches a preset differential-pressure reference value.

In accordance with a third aspect of the present invention, there is provided a method of controlling a solenoid of a swash plate type variable displacement compressor having a swash plate and a cone clutch. The swash plate is inclinable at a variable inclination angle. The cone clutch is engageable by energization of the solenoid. The method includes the steps of: detecting a first pressure of a refrigerant or a first pressure that reflects the first pressure of the refrigerant, when the swash plate is at a minimum inclination angle position; starting passing an electric current through the solenoid; detecting a second pressure of the refrigerant or a second element that reflects the second pressure of the refrigerant after the step of starting passing the electric current through the solenoid; and stopping passing the electric current through the solenoid if a value of change between the first pressure and the second pressure reaches a preset reference value or if a value of change between the first element and the second element reaches a preset reference value.

Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

## BRIEF DESCRIPTION OF THE DRAWINGS

The invention together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

FIG. **1** is a longitudinal sectional view showing a variable displacement compressor according to a first embodiment of the present invention and its related devices;

FIG. **2** is a partially enlarged sectional view of a swash plate of the variable displacement compressor of FIG. **1**, showing a state where the swash plate is placed at the maximum inclination angle position;

FIG. **3** is a partially enlarged sectional view of the swash plate of the variable displacement compressor of FIG. **1**, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **4** is a partially enlarged sectional plan view showing a hinge mechanism of the variable displacement compressor of FIG. **1**;

FIG. **5** is a partially enlarged sectional view showing a stop of the variable displacement compressor of FIG. **1**;

FIG. **6** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a second embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **7** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a third embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **8** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a fourth embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **9** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a fifth embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **10** is a partially enlarged view showing a first lubrication groove of the variable displacement compressor of FIG. **9**;

FIG. **11** is a cross sectional view showing the variable displacement compressor as taken along the line A-A of FIG. **9**;

FIG. **12** is a cross sectional view similar to FIG. **11**, but showing a variable displacement compressor according to a sixth embodiment of the present invention;

FIG. **13** is a sectional view showing the variable displacement compressor as taken along the line B-B of FIG. **12**;

FIG. **14** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a seventh embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **15** is a graph illustrating relationship of the gap and the electromagnetic force;

FIG. **16** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to an eighth embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **17** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a ninth embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **18** is a cross sectional view showing the variable displacement compressor as taken along the line C-C of FIG. **17**;

FIG. **19** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a tenth embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **20** is a cross sectional view showing the variable displacement compressor as taken along the line D-D of FIG. **19**;

FIG. **21** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to an eleventh embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **22** is a graph illustrating a change of top clearance of a piston of the variable displacement compressor of FIG. **21**;

FIG. **23** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a twelfth embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **24** is a cross sectional view showing the variable displacement compressor as taken along the line E-E of FIG. **23**;

FIG. **25** is a longitudinal sectional view showing a variable displacement compressor according to a thirteenth embodiment of the present invention and its related devices;

FIG. **26** is a flowchart illustrating the operation of the variable displacement compressor of FIG. **25**;

FIG. **27** is a longitudinal sectional view showing a variable displacement compressor according to a fourteenth embodiment of the present invention and its related devices;

FIG. **28** is a flowchart illustrating the operation of the variable displacement compressor of FIG. **27**;

FIG. **29** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a fifteenth embodiment of the present invention, showing a state where the swash plate is at the maximum inclination angle position;

FIG. **30** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a sixteenth embodiment of the present invention, showing a state where the swash plate is placed at the maximum inclination angle position;

FIG. **31** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a seventeenth embodiment of the present invention, showing a state where the swash plate is placed at the maximum inclination angle position;

FIG. **32** is a cross sectional view showing the variable displacement compressor as taken along the line F-F of FIG. **31**;

FIG. **33** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to an eighteenth embodiment of the present invention, showing a state where the swash plate is placed at the maximum inclination angle position;

FIG. **34** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a nineteenth embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **35** is a partially enlarged sectional view of the swash plate of the variable displacement compressor of FIG. **34**, showing a state where the swash plate is placed at the maximum inclination angle position;

FIG. **36** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a twentieth embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position;

FIG. **37** is a partially enlarged sectional view of the swash plate of the variable displacement compressor of FIG. **36**, showing a state where the swash plate is placed at the maximum inclination angle position;

FIG. **38** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a twenty-first embodiment of the present invention, showing a state where the swash plate is placed at the maximum inclination angle position; and

FIG. **39** is a partially enlarged sectional view of a swash plate of a variable displacement compressor according to a modification of the third embodiment of the present invention, showing a state where the swash plate is placed at the minimum inclination angle position.

## DETAILED DESCRIPTION OF THE EMBODIMENTS

The following will describe the first embodiment of the present invention with reference to FIGS. **1** through **5**. Referring to FIG. **1** showing the variable displacement compressor in longitudinal sectional view, the variable displacement compressor is generally designated by reference numeral **10** and includes a cylinder block **11**. The left-hand side and the right-hand side of the variable displacement compressor **10** as seen in FIG. **1** correspond to the front and the rear of the variable displacement compressor **10**, respectively. A front housing **12** is joined to the cylinder block **11** at the front end thereof. A rear housing **13** is joined to the cylinder block **11** at the rear end thereof via a port plate **14**, a suction valve plate **15**, a discharge valve plate **16** and a retainer plate **17**. The cylinder block **11**, the front housing **12** and the rear housing **13** cooperate to form a compressor housing of the variable displacement compressor **10**.

The front housing **12** and the cylinder block **11** cooperate to form a crank chamber **121** and rotatably support a rotary shaft **18** via radial bearings **19** and **20**, respectively. The rotary shaft **18** extends forward of the crank chamber **121** for receiving driving force from a vehicle engine (not shown). A shaft seal device **21** of a lip seal type is interposed between the front housing **12** and the rotary shaft **18** for preventing refrigerant from leaking along the peripheral surface of the rotary shaft **18** out of the crank chamber **121**.

A first rotor **22** is fixed on the rotary shaft **18** for rotation therewith. The first rotor **22** is formed in an annular shape to have an axial hole **221** through which the rotary shaft **18** is fixedly fitted. A swash plate **23** is supported on the rotary shaft **18** so as to be slidable along and inclinable with respect to the axis **181** of the rotary shaft **18**. The swash plate **23** is disposed in the crank chamber **121**. A second rotor **24** is interposed between the first rotor **22** and the swash plate **23**.

The first rotor **22** has a male cone portion **25** fixed on the rotary shaft **18** and a pressure receiving portion **26** that extends radially outward from the outer periphery of the male cone portion **25**. The pressure receiving portion **26** has a shape of an annular plate. The male cone portion **25** tapers toward the swash plate **23** and has a conical surface **251** that surrounds the axis **181** of the rotary shaft **18**. The axis of the male cone portion **25** coincides with the axis **181**.

The second rotor **24** has a female cone portion **27** that is connectable to and disconnectable from the male cone portion **25** of the first rotor **22**, and an attraction receiving portion **28** that extends radially outward from the female cone portion **27** and has a shape of an annular plate. The outside diameter of the attraction receiving portion **28** of the second rotor **24** is made larger than that of the pressure receiving portion **26** of the first rotor **22**. As viewed in the direction of the axis **181**, the outer periphery of the pressure receiving portion **26** is located radially inward of the outer periphery of the attraction receiving portion **28**.

The female cone portion **27** of the second rotor **24** tapers toward the swash plate **23** and has a conical surface **271** that surrounds the axis **181** of the rotary shaft **18**. The axis of the female cone portion **27** coincides with the axis **181**. The second rotor **24** is slidable along the rotary shaft **18** to move the conical surface **271** of the female cone portion **27** into and out of a joint contact with the conical surface **251** of the male cone portion **25**. Thus, the male cone portion **25** and the female cone portion **27** cooperate to form a cone clutch K. The second rotor **24** is made of a magnetic material.

Referring to FIGS. **2** and **3**, a thrust bearing **29** is interposed between the pressure receiving portion **26** of the first rotor **22** and the front housing **12**, and a thrust bearing **30** is interposed between the pressure receiving portion **26** and the female cone portion **27**. The thrust bearing **30** is a rolling bearing. A disc spring **31** is interposed between the thrust bearing **30** and the pressure receiving portion **26** as a spring member that serves as a spacer. The thrust bearing **30** serves to reduce the sliding resistance between the disc spring **31** and the female cone portion **27**. The disc spring **31** and the thrust bearing **30** are disposed surrounding the conical surface **251** of the male cone portion **25**. The disc spring **31** presses the thrust bearing **30** against the surface **272** of the female cone portion **27** that faces the pressure receiving portion **26**.

An annular solenoid **32** is mounted on the inner surface of the front housing **12** and disposed so as to surround the rotary shaft **18**. The solenoid **32** has a coil **33** and a coil holder **34** that holds the coil **33**. The coil holder **34** is made of a magnetic material and opened toward the attraction receiving portion **28** of the second rotor **24**. When an electric current is passed through the coil **33**, the attraction receiving portion **28** of the second rotor **24** receives attraction force (or electromagnetic force) produced by the solenoid **32**. The solenoid **32**, the first rotor **22** and the second rotor **24** cooperate to form an electromagnetic clutch that is incorporated in the compressor housing.

Referring to FIG. **4**, a pair of projections **37** and **38** extends from the second rotor **24** toward the swash plate **23**, and a pair of arms **35** and **36** extends from the swash plate **23** toward the second rotor **24**. The paired arms **35** and **36** are inserted in a recess **39** formed between the paired projections **37** and **38**. The paired arms **35** and **36** are sandwiched in between the paired projections **37** and **38** and movable in the recess **39**. The innermost part of the recess **39** is formed as a cam surface **391** on which the distal ends **351** and **361** of the arms **35** and **36** are slidable. The paired arms **35** and **36** sandwiched in between the paired projections **37** and **38** work in cooperation with the cam surface **391** in such a way that the swash plate **23** is inclinable with respect to the axis **181** of the rotary shaft **18** and rotatable integrally with the rotary shaft **18**. The paired arms **35**, **36** and the paired projections **37**, **38** cooperate to form a hinge mechanism **40** that allows the swash plate **23** to incline relative to the second rotor **24** and also allows torque transmission from the second rotor **24** to the swash plate **23**.

As shown in FIGS. **2** and **3**, θ represents an inclination angle of the swash plate **23** that is made between the central axis **231** of the swash plate **23** and the axis **181** of the rotary shaft **18**. In the present embodiment, when the inclination angle θ of the swash plate **23** is minimum as shown in FIG. **3**, the lines N (only one line being shown) that pass through the contact points T (only one contact point being shown) between the paired arms **35**, **36** and the cam surface **391** and are normal to the cam surface **391** are set so as to extend inward of the inner peripheries of the thrust bearing **30** and the disc spring **31**.

When the central part of the swash plate **23** is moved toward the second rotor **24** (or in forward direction), the inclination angle θ of the swash plate **23** increases. The maximum inclination angle of the swash plate **23** is determined by the contact between the second rotor **24** and the swash plate **23**.

A return spring **60** is interposed between the swash plate **23** and the cylinder block **11** so as to surround the rotary shaft **18** for urging the swash plate **23** in the direction that causes the inclination angle θ of the swash plate **23** to be increased. The minimum inclination angle of the swash plate **23** is determined by the contact of the swash plate **23** with the front end of the return spring **60**. The swash plate **23** shown in FIG. **1** (by the solid line) and FIG. **2** is placed at the maximum inclination angle position. The swash plate **23** shown in FIG. **1** (by the chain double-dashed line) and FIG. **3** is placed at the minimum inclination angle position. The minimum inclination angle of the swash plate **23** is set slightly larger than 0°.

As shown in FIG. **1**, an inclination-angle reduction spring **41** is interposed between the male cone portion **25** of the first rotor **22** and the swash plate **23** so as to surround the rotary shaft **18**. A stop **42** formed by an annular plain bearing is interposed between the inclination-angle reduction spring **41** and the male cone portion **25**. The inclination-angle reduction spring **41** urges the swash plate **23** in the direction that causes the inclination angle θ of the swash plate **23** to be decreased.

The combined spring characteristics of the inclination-angle reduction spring **41** and the return spring **60** is set so that the swash plate **23** is placed at the minimum inclination angle position when the pressure in the variable displacement compressor **10** is uniform and the swash plate **23** is not rotated.

Referring to FIG. **5**, the stop **42** has an inner ring portion **43** and an outer ring portion **44**. The inner ring portion **43** is in contact with the first rotor **22** and the inclination-angle reduction spring **41**, and the outer ring portion **44** is connectable to and disconnectable from the second rotor **24**, as seen from FIGS. **2** and **3**. The front surface **431** of the inner ring portion **43** is pressed against the end surface **252** of the male cone portion **25** of the first rotor **22** by the spring force of the inclination-angle reduction spring **41**. The front surface **441** of the outer ring portion **44** is located to face the end surface **273** of the female cone portion **27** of the second rotor **24**. The front surface **441** of the outer ring portion **44** recedes from the front surface **431** of the inner ring portion **43** toward the swash plate **23**. That is, the front surface **441** is spaced from the end surface **252** of the male cone portion **25** toward the swash plate **23**.

Referring back to FIG. **1**, the cylinder block **11** has therethrough a plurality of cylinder bores **111** each receiving therein a piston **45**. Each piston **45** is engaged with the outer periphery of the swash plate **23** via a pair of shoes **46**. Rotation of the swash plate **23** is converted into the reciprocating motion of the pistons **45** via the pairs of shoes **46**. Thus, the pistons **45** are reciprocable in the respective cylinder bores **111**. The length of the stroke of each piston **45** moving its cylinder bore **111** is variable depending on the inclination angle of the swash plate **23**.

The rear housing **13** has therein a suction chamber **131** and a discharge chamber **132** that form a suction pressure region and a discharge pressure region of the compressor **10**, respectively. Each of the port plate **14**, the discharge valve plate **16** and the retainer plate **17** has therethrough a plurality of suction ports **47**. Each of the port plate **14** and the suction valve plate **15** has therethrough a plurality of discharge ports **48**. The suction valve plate **15** has a plurality of suction valves **151**, and the discharge valve plate **16** has a plurality of discharge valves **161**. A compression chamber **112** is formed in each cylinder bore **111** between the suction valve plate **15** and the piston **45** in the cylinder bore **111**.

When the piston **45** is moved forward (or moved leftward as seen in FIG. **1**), refrigerant in the suction chamber **131** flows into its compression chamber **112** while pushing away the suction valve **151** thereby to open the suction port **47**. When the piston **45** is moved rearward (or moved rightward as seen in FIG. **1**), the refrigerant is compressed in the compression chamber **112** and discharged into the discharge chamber **132** while pushing away the discharge valve **161** thereby to open the discharge port **48**. The opening of the discharge valve **161** is restricted by the contact of the discharge valve **161** with the retainer **171** formed on the retainer plate **17**.

When the pressure in the crank chamber **121** decreases, the inclination angle of the swash plate **23** is increased to increase the displacement of the variable displacement compressor **10**. When the pressure in the crank chamber **121** increases, on the other hand, the inclination angle of the swash plate **23** is decreased to decrease the displacement of the variable displacement compressor **10**. The suction chamber **131** and the discharge chamber **132** are connected via an external refrigerant circuit **49** that includes a condenser **50**, an expansion valve **51** and an evaporator **52**. The condenser **50** absorbs heat from the refrigerant flowing therethrough, and the evaporator **52** transfers the surrounding heat to the refrigerant flowing in the evaporator **52**. A check valve **53** is located between the discharge chamber **132** and the external refrigerant circuit **49**. The refrigerant in the discharge chamber **132** flows through the check valve **53** into the external refrigerant circuit **49**.

The reaction force developed when the refrigerant is discharged from the compression chamber **112** is received by the front housing **12** via the cylinder bore **111**, the piston **45**, the pair of shoes **46**, the swash plate **23**, the hinge mechanism **40**, the second rotor **24**, the cone clutch K, the first rotor **22** and the thrust bearing **29**.

The discharge chamber **132** and the crank chamber **121** are connected via a supply passage **54**. The crank chamber **121** and the suction chamber **131** are connected via a bleed passage **55**. An electromagnetically-operated displacement control valve **56** is connected in the supply passage **54**. A control computer C is connected to the displacement control valve **56** for controlling passage of an electric current with duty ratio through the displacement control valve **56**. The control computer C is connected to an air-conditioner operation switch **57**. The control computer C passes an electric current through the displacement control valve **56** when the air-conditioner operation switch **57** is ON. The control computer C stops passing the electric current through the displacement control valve **56** when the air-conditioner operation switch **57** is OFF. A room temperature setting device **58** and a room temperature sensor **59** are connected to the control computer C by signals. When the air-conditioner operation switch **57** is ON, the control computer C controls passage of the electric current through the displacement control valve **56** in accordance with the difference between a target room temperature set by the room temperature setting device **58** and a room temperature sensed by the room temperature sensor **59**. The opening of the displacement control valve **56** decreases as the duty ratio increases.

The following will describe the operation of the first embodiment. When the swash plate **23** is placed at the minimum inclination angle position and the cone clutch K is disengaged, as shown in FIG. **3**, the passage of the electric current through the displacement control valve **56** is stopped and the opening of the displacement control valve **56** is maximum. When the swash plate **23** is placed at the minimum inclination angle position, there is slight differential pressure between the compression chambers **112** and the crank chamber **121**, so that the reaction force received by the swash plate **23** due to the differential pressure is relatively small. Therefore, the second rotor **24** is located in the position where the female cone portion **27** is in contact with the stop **42** by the spring force of the disc spring **31**.

When the passage of the electric current through the displacement control valve **56** is started, energization of the solenoid **32** is also started. When the energization of the solenoid **32** is started, the attraction receiving portion **28** of the second rotor **24** is attracted toward the solenoid **32** against the spring force of the disc spring **31**, so that the conical surface **271** of the female cone portion **27** comes in contact with the conical surface **251** of the male cone portion **25**. That is, the cone clutch K is shifted from the disengaged state to the engaged state. When the cone clutch K is engaged, the rotation of the first rotor **22** is transmitted to the second rotor **24** via the cone clutch K thereby to rotate the second rotor **24** and the swash plate **23** integrally with the first rotor **22**. Energization of the solenoid **32** is stopped when it is considered that a time taken to shift the cone clutch K from the disengaged state to the engaged state has passed since the start of the energization of the solenoid **32**.

When the passage of the electric current through the displacement control valve **56** is started, the opening of the displacement control valve **56** decreases. In this case, the cone clutch K is engaged thereby to rotate the swash plate **23**, so that the refrigerant is discharged from the compression chambers **112** into the discharge chamber **132**. Thus, the inclination angle of the swash plate **23** increases. With an increase of the inclination angle of the swash plate **23** from the minimum inclination angle, the discharge pressure also increases. When the discharge pressure increases, the check valve **53** is opened thereby to allow the refrigerant in the discharge chamber **132** to flow into the external refrigerant circuit **49**. The refrigerant flowed into the external refrigerant circuit **49** returns to the suction chamber **131**.

When the current value supplied to the displacement control valve **56** is increased, the opening of the displacement control valve **56** is decreased thereby to decrease the refrigerant supplied from the discharge chamber **132** to the crank chamber **121**. Since part of the refrigerant in the crank chamber **121** flows into the suction chamber **131** via the bleed passage **55**, the pressure in the crank chamber **121** decreases with a decrease of the supply of the refrigerant, so that the inclination angle of the swash plate **23** is increased and hence the displacement of variable displacement compressor **10** is increased. When the current value supplied to the displacement control valve **56** is decreased, on the other hand, the opening of the displacement control valve **56** is increased thereby to increase the refrigerant supplied from the discharge chamber **132** into the crank chamber **121**. Therefore, the pressure in the crank chamber **121** increases, so that the inclination angle of the swash plate **23** is decreased and hence the displacement of variable displacement compressor **10** is decreased.

When the duty ratio becomes zero, or when the energization of the displacement control valve **56** is stopped, the opening of the displacement control valve **56** becomes maximum. The second rotor **24** and the swash plate **23** are then located in the position shown in FIG. **3** by the spring force of the disc spring **31**. When the swash plate **23** stops rotating, the check valve **53** is closed thereby to stop the refrigerant from flowing through the external refrigerant circuit **49**.

The following will describe the advantageous effects of the first embodiment.

(1) The electromagnetic clutch including the solenoid **32** and the cone clutch K is disengaged when the inclination angle of the swash plate **23** is minimum, so that the second rotor **24** is then disconnected from the first rotor **22**. With the swash plate **23** placed at the minimum inclination angle position, therefore, the swash plate **23** is free from integral rotation with the second rotor **24**. Thus, mechanical loss of the variable displacement compressor **10** is reduced.

(2) While the inclination angle of the swash plate **23** is increased from the disengaged state of the cone clutch K, the electromagnetic clutch is engaged temporarily. When the electromagnetic clutch is engaged, the first rotor **22** and the second rotor **24** are rotated integrally thereby to rotate the swash plate **23** with the second rotor **24** integrally. The inclination angle of the swash plate **23** is increased and the reaction force developed due to the discharging of refrigerant is also increased, so that the engaged state of the cone clutch K is kept though the energization of the solenoid **32** is then

stopped. Since the electromagnetic clutch is engaged only temporarily, the power consumption of the compressor **10** is reduced extremely.

(3) The first rotor **22** is located radially inward of the annular solenoid **32** as viewed in the direction of the axis **181** of the rotary shaft **18**. The structure wherein the inside diameter of the solenoid **32** is made larger than the outside diameter of the first rotor **22** is advantageous in that the diameter of the solenoid **32** is increased thereby to enhance the electromagnetic force.

(4) If the attraction receiving portion **28** of the second rotor **24** is spaced too far from the solenoid **32**, the electromagnetic force of the solenoid **32** acting on the attraction receiving portion **28** is reduced, which makes it difficult to engage the cone clutch K. The stop **42** regulates the distance in the direction of the axis **181** between the solenoid **32** and the attraction receiving portion **28**, or the distance in the direction of the axis **181** between the first rotor **22** and the second rotor **24**, in such a way that the electromagnetic force of the solenoid **32** has a magnitude that is strong enough for the attraction receiving portion **28** to be attracted to the solenoid **32**.

(5) The inclination-angle reduction spring **41** is simple in structure, but effective to hold the stop **42** in place.

(6) When the inclination angle of the swash plate **23** is minimum, the end surface **273** of the female cone portion **27** of the second rotor **24** is in contact with the outer ring portion **44** of the stop **42**. The stop **42** formed by the plain bearing prevents the second rotor **24** from rotating with the swash plate **23** when the swash plate **23** is placed at the minimum inclination angle position.

(7) When the swash plate **23** is placed into the minimum inclination angle position, the spring force of the disc spring **31** causes the conical surfaces **251** and **271** of the first and second rotors **22** and **24** to be spaced from each other thereby to shift the cone clutch K from the engaged state to the disengaged state. When the swash plate **23** is at the minimum inclination angle position, therefore, the second rotor **24** becomes free from rotation with the first rotor **22**. In order to shift the cone clutch K from the disengaged state to the engaged state, the electromagnetic force of the solenoid **32** acting on the attraction receiving portion **28** of the disengaged cone clutch K should be strong enough, or, alternatively, the distance in the direction of the axis **181** between the solenoid **32** and the attraction receiving portion **28** may be reduced. In order to shift the cone clutch K from the engaged state to the disengaged state, on the other hand, the force that pulls the conical surfaces **251** and **271** away from each other should be strong enough.

The use of the disc spring **31** having a small amount of elastic deformation is advantageous in that the distance in the direction of the axis **181** between the solenoid **32** and the attraction receiving portion **28** may be reduced when the cone clutch K is disengaged and also that the spring force may be increased when the cone clutch K is engaged.

(8) While the cone clutch K is disengaged, there is fear that the second rotor **24** may be inclined by the arms **35** and **36** then pressing against the cam surface **391**. More specifically, the second rotor **24** may be inclined in the direction that causes the arms **35** and **36** to be moved toward the solenoid **32**, or in the direction that causes the upper side of the attraction receiving portion **28** of the second rotor **24** to be moved toward the solenoid **32** as seen in FIG. **3**. Such an inclination may cause the contact between the solenoid **32** and the attraction receiving portion **28**, which produces abrasion powder in a region of the contact.

It is so set that when the swash plate **23** is placed at the minimum inclination angle position, the normal lines N (only one line being shown) to the cam surface **391** at the contact points T (only one point being shown) between the paired arms **35**, **36** and the cam surface **391** pass inward of the inner peripheries of the thrust bearing **30** and the disc spring **31**. Such setting prevents the inclination of the second rotor **24** caused by pressing of the arms **35** and **36** against the cam surface **391**.



The following will describe the second embodiment of the present invention with reference to FIG. **6**. The same reference numerals are used for the common elements or components in the first and second embodiments, and the description of such elements or components for the second embodiment will be omitted.

Referring to FIG. **6**, the first rotor **22**A corresponding to the first rotor **22** of the first embodiment has a cylindrical guide portion **61** and a male cone portion **25**A that is located radially outward of the cylindrical guide portion **61** and serves as a pressure receiving portion. The cylindrical guide portion **61** is fixed on the rotary shaft **18**. The disc spring **31** and the thrust bearing **30** are disposed surrounding the cylindrical guide portion **61**.

The second rotor **24**A corresponding to the second rotor **24** of the first embodiment receives therein the cylindrical guide portion **61** of the first rotor **22**A so as to be rotatable relative to and slidable on the cylindrical guide portion **61**. The conical surface **271** of the female cone portion **27** of the second rotor **24**A surrounds the disc spring **31** and the thrust bearing **30**. The second rotor **24**A has a radially inner peripheral surface **241** that serves as a cylindrical surface. The inner peripheral surface **241** and a radially outer peripheral surface **611** of the cylindrical guide portion **61** are in contact with each other. When the solenoid **32** is energized with the swash plate **23** placed at the minimum inclination angle position, the second rotor **24**A is moved in the direction of the axis **181** while being guided by the outer peripheral surface **611** of the cylindrical guide portion **61**. The cylindrical guide portion **61** and the inner peripheral surface **241** cooperate to form a guide that guides the second rotor **24**A so as to be rotatable relative to and slidable on the cylindrical guide portion **61**.

In the second embodiment, the same advantageous effects as those in the first embodiment are obtained and the following additional effects are also obtained. In the first embodiment, when the cone clutch K is shifted from the disengaged state to the engaged state, the second rotor **24** may be inclined relative to the axis **181**. If the second rotor **24** is thus inclined, the surface **281** of the attraction receiving portion **28** of the second rotor **24** that faces the solenoid **32** is not kept parallel to the attraction surface **321** of the solenoid **32**, so that the electromagnetic force of the solenoid **32** fails to act on the attraction receiving portion **28** of the second rotor **24** along the circumference of the attraction receiving portion **28** uniformly. This causes the cone clutch K to be displaced from the disengaged state to the engaged state with the second rotor **24** inclined relative to the axis **181**. Thus, the attraction receiving portion **28** of the second rotor **24** may come into contact with the solenoid **32**, which produces abrasion powder in the region of contact between the non-rotating solenoid **32** and the rotating second rotor **24**.

In the second embodiment, the second rotor **24**A which is constantly supported by the cylindrical guide portion **61** of the first rotor **22**A will not be inclined relative to the axis **181**. Therefore, the compressor **10** according to the second embodiment is free from the problem associated with the abrasion powder caused by the inclination of the second rotor **24**A.

The following will describe the third embodiment of the present invention with reference to FIG. **7**. The same reference numerals are used for the common elements or components in the second and third embodiments, and the description of such elements or components for the third embodiment will be omitted.

Referring to FIG. **7**, a rolling bearing **62** is interposed as a radial bearing between the outer peripheral surface **611** of the cylindrical guide portion **61** and the inner peripheral surface **241** of the second rotor **24**A. The rolling bearing **62** serves to smoothen the relative rotation and sliding motion between the first rotor **22**A and the second rotor **24**A.

The following will describe the fourth embodiment of the present invention with reference to FIG. **8**. The same reference numerals are used for the common elements or components in the second and fourth embodiments, and the description of such elements or components for the fourth embodiment will be omitted.

Referring to FIG. **8**, a second rotor **24**A has a cylindrical guide portion **63** at a position that is radially outward of the male cone portion **25**A and surrounds the first rotor **22**A. The first rotor **22**A is fitted in the cylindrical guide portion **63** of the second rotor **24**A. The first rotor **22** has a radially outer peripheral surface **222** of the first rotor **22**A that serves as a cylindrical surface. The outer peripheral surface **222** and a radially inner peripheral surface **631** of the cylindrical guide portion **63** of the second rotor **24**A are in contact with each other. The cylindrical guide portion **63** and the outer peripheral surface **222** cooperate to form a guide that guides the second rotor **24**A rotatably and slidably relative to the male cone portion **25**A.

Although the cylindrical guide portion **63** plays a role that is similar to the cylindrical guide portion **61**, the cylindrical guide portion **63** having a larger inside diameter than the outside diameter of the cylindrical guide portion **61** is more effective in preventing the inclination of the second rotor **24**A than the cylindrical guide portion **61** in the second embodiment. The structure wherein the second rotor **24**A is guided at the peripheral surfaces **241** and **631** is particularly effective in preventing the inclination of the second rotor **24**A and also in smoothening the sliding motion of the second rotor **24**A.

The following will describe the fifth embodiment of the present invention with reference to FIGS. **9** through **11**. The same reference numerals are used for the common elements or components in the first and fifth embodiments, and the description of such elements or components for the fifth embodiment will be omitted.

Referring to FIGS. **9** and **10**, an annular coil cover **64** is provided on the surface of the coil **33** that faces the attraction receiving portion **28** of the second rotor **24** (or in the opening of the coil holder **34**). The coil cover **64** is made of a resin for sealing the coil **33** in the coil holder **34**.

Referring to FIGS. **9** and **11**, one first lubrication groove **65** and two second lubrication grooves **66** are formed radially in the surface of the coil cover **64** and the annular surface **641** of the coil holder **34** that face the attraction receiving portion **28**. The first lubrication groove **65** is located below the axis **181** and the second lubrication grooves **66** are located above the axis **181**. Specifically, the first lubrication groove **65** is located at the bottom of the coil cover **64**. The first lubrication groove **65** and the second lubrication grooves **66** are formed radially across the coil cover **64** and the annular surface **641** of the coil holder **34**. The second lubrication groove **66** communicates at the inner periphery of the annular surface **641** with the radially inner region of the solenoid **32**.

When the swash plate **23** is placed at the minimum inclination angle position, lubricating oil is accumulated in the bottom of the crank chamber **121** and flows into the first lubrication groove **65**. While the cone clutch K is in the disengaged state, the male cone portion **25** and the female cone portion **27** may come in contact with the each other occasionally, so that the second rotor **24** is rotated with the first rotor **22**. Such rotation of the second rotor **24** with the first rotor **22** causes the lubricating oil in the first lubrication groove **65** to be pulled up as oil film through the space between the coil cover **64** and the attraction receiving portion **28**.

In order to pull up the lubricating oil, it is necessary to rotate the second rotor **24** with the first rotor **22**. In the present embodiment, the diameter of the return spring **60** is made larger than that of the inclination-angle reduction spring **41** as shown in FIG. **9**. That is, the point of action of the return spring **60** that acts on the swash plate **23** (or the starting point Q1 of the arrow Q that represents the direction of action) is located radially outward of the point of action of the inclination-angle reduction spring **41** on the swash plate **23** (or the starting point R1 of the arrow R that represents the direction of action). In this structure, the swash plate **23** at the minimum inclination angle position is subjected to a force acting in a counterclockwise direction as seen in FIG. **9** by the action of the return spring **60** and the action of the inclination-angle reduction spring **41**, so that the cam surface **391** is pressed by the arms **35** and **36**. The actions of the arms **35** and **36** pressing against the cam surface **391** increase the tendency of the conical surfaces **271** and **251** to contact with each other and hence the tendency of the second rotor **24** to be rotated with the first rotor **22** is enhanced.

A part of the lubricating oil pulled up is flowed into the second lubrication grooves **66** and then supplied to the thrust bearing **30** that is located radially inward of the solenoid **32**.

A part of the lubricating oil that is attached to a rotary member, such as the first rotor **22**, the disc spring **31** or the thrust bearing **30**, flows along the inner peripheral surface of the solenoid **32** into the space between the first rotor **22** and the front housing **12** by centrifugal force. The thrust bearing **29**, the radial bearing **19** and the shaft seal device **21** are lubricated by the lubricating oil flowed into the space between the first rotor **22** and the front housing **12**.

When the swash plate **23** is placed at the minimum inclination angle position, the rotary shaft **18** and the first rotor **22** are rotated and hence the thrust bearing **29**, the radial bearing **19** and the shaft seal device **21** need to be lubricated. The lubricating oil in the first lubrication groove **65** and the second lubrication groove **66** is supplied to the thrust bearing **29**, the radial bearing **19** and the shaft seal device **21** for lubrication thereof. Thus, the thrust bearing **29**, the radial bearing **19** and the shaft seal device **21** are lubricated appropriately when the swash plate **23** is placed at the minimum inclination angle position.

In the structure according to the fifth embodiment, the first lubrication groove **65** located at the bottom of the coil cover **64** is likely to be immersed in the lubricating oil accumulated in the bottom of the crank chamber **121**. That is, the present embodiment wherein the first lubrication groove **65** is located at the bottom of the coil cover **64** is effective in pulling the lubricating oil into the first lubrication groove **65**.

The following will describe the sixth embodiment of the present invention with reference to FIGS. **12** and **13**. The same reference numerals are used for the common elements or components in the fifth and sixth embodiments, and the description of such elements or components for the sixth embodiment will be omitted.

Referring to FIGS. **12** and **13**, a first annular lubrication groove **67** and a second annular lubrication groove **68** are formed in the annular surface **641** of the coil holder **34** at positions that are radially inward and outward of the annular surface **641**, respectively, so as to extend along the circumferential direction of the coil cover **64**. The first annular lubrication groove **67** is located radially inward of the second annular lubrication groove **68**. The first lubrication groove **65** and the second lubrication grooves **66** are formed radially across the first annular lubrication groove **67** and the second annular lubrication groove **68**. Each of the first lubrication groove **65** and the second lubrication grooves **66** communicates at the inner periphery of the coil holder **34** with the radially inner region of the solenoid **32** and at the outer periphery of the coil holder **34** with the radially outer region of the solenoid **32**. The first lubrication groove **65** and the second lubrication grooves **66** are connected to the first annular lubrication groove **67** and the second annular lubrication groove **68**.

The first annular lubrication groove **67** and the second annular lubrication groove **68** serve to prevent the lubricating oil that is pulled upward from the lower part of the solenoid **32** from leaking radially outward due to the centrifugal force, thereby guiding the lubricating oil toward the upper part of the solenoid **32** and lubricating the thrust bearings **30**, **29**, the radial bearing **19** and the shaft seal device **21** successfully.

The following will describe the seventh embodiment of the present invention with reference to FIGS. **14** and **15**. The same reference numerals are used for the common elements or components in the first and seventh embodiments, and the description of such elements or components for the seventh embodiment will be omitted.

Referring to FIG. **14**, the coil holder **34** has a projection extending from the radially outer annular end of the coil holder **34** toward the attraction receiving portion **28** of the second rotor **24** and having a surface **69** that is tapered away from the second rotor **24**. The attraction receiving portion **28** of the second rotor **24** has a radially outer portion (or annular portion) having a surface **70** that is tapered toward the solenoid **32**. The tapered surface **70** faces the tapered surface **69** so as to be complementary to the tapered surface **69**. A gap L**1** is formed between the tapered surface **69** of the solenoid **32** and its complementary tapered surface **70** of the attraction receiving portion **28** of the second rotor **24** when the cone clutch K is in the disengaged state. L**2** in FIG. **14** represents a gap between the solenoid **32** and the attraction receiving portion **28** of the second rotor **24** when the cone clutch K is in the disengaged state, as measured in the direction parallel to the axis **181**. As apparent from the drawing, L**1** is smaller than L**2**. The provision of the tapered surfaces **69** and **70** that provide the smaller gap L**1** increases the electromagnetic force of the solenoid **32** acting on the attraction receiving portion **28**.

Referring to FIG. **15**, the horizontal axis and vertical axis of the graph represent the magnitude of the gap and the electromagnetic force. The curve D shows an example of the change of the electromagnetic force produced when the tapered surfaces **69** and **70** are not provided. The curve E shows an example of the change of the electromagnetic force produced when the tapered surfaces **69** and **70** are provided. The straight line F shows an example of the change of the spring force of the disc spring **31**. A disc spring such as **31** having a larger spring force may be used when the tapered surfaces **69** and **70** are provided than when no such tapered surfaces are provided. That is, although the disc spring **31** having a large spring force is used, shifting of the electromagnetic clutch from the disengaged state to the engaged state and vice versa can be done steadily.

As described above, while the cone clutch K is in the disengaged state, there is fear that the second rotor **24** may be inclined by the arms **35** and **36** then pressing against the cam surface **391**. If the second rotor **24** is inclined, the gap between the solenoid **32** and the attraction receiving portion **28** along the circumference of the attraction receiving portion **28** becomes non-uniform. The gap between the solenoid **32** and the attraction receiving portion **28** is minimum at a position adjacent to the hinge mechanism **40** in the circumferential direction of the second rotor **24**. Such non-uniformity of the gap makes non-uniform the electromagnetic force of the solenoid **32** acting on the attraction receiving portion **28** in the circumferential direction of the attraction receiving portion **28**. In this case, the electromagnetic force acting on the attraction receiving portion **28** at a position adjacent to the hinge mechanism **40** is maximum. When the solenoid **32** is energized with the second rotor **24** thus inclined, the above-described non-uniform gap is further increased (or the second rotor **24** is further inclined).

As is apparent from the curves E and D in FIG. **15**, the rate of the change of the electromagnetic force relative to the change of the gap is smaller when the tapered surfaces **69** and **70** are provided than when the same tapered surfaces are not provided. That is, the provision of the tapered surfaces **69** and **70** helps to reduce the non-uniformity in the circumferential direction of the attraction receiving portion **28**, of the electromagnetic force of the solenoid **32** acting on the attraction receiving portion **28**. This helps to reduce the inclination of the second rotor **24** occurring in energizing the solenoid **32**.

The following will describe the eighth embodiment of the present invention with reference to FIG. **16**. The same reference numerals are used for the common elements or components in the seventh and eighth embodiments, and the description of such elements or components for the eighth embodiment will be omitted.

Referring to FIG. **16**, the coil holder **34** has at the radially inner annular end adjacent to the second rotor **24** a surface **71** that is tapered away from the attraction receiving portion **28** of the second rotor **24**. The attraction receiving portion **28** of the second rotor **24** has a radially inner portion (or annular portion) having a surface **72** that is tapered toward the coil holder **34**. The tapered surface **72** faces the tapered surface **71** so as to be complementary to the tapered surface **71**.

The eighth embodiment has substantially the same advantageous effects as the seventh embodiment. In addition, the eighth embodiment wherein the tapered surfaces **71** and **72** are added causes more the electromagnetic force of the solenoid **32** acting on the attraction receiving portion **28** than the seventh embodiment.

The following will describe the ninth embodiment of the present invention with reference to FIGS. **17** and **18**. The same reference numerals are used for the common elements or components in the first and ninth embodiments, and the description of such elements or components for the ninth embodiment will be omitted.

Referring to FIGS. **17** and **18**, the second rotor **24** has therethrough a plurality of arched voids **73** that are formed in a concentric manner. The voids **73** are located radially inward of the solenoid **32** as viewed in the direction of the axis **181**. The voids **73** are a flux barrier located radially inward of the attraction receiving portion **28**. That is, the voids **73** serve to reduce flux leakage from the attraction receiving portion **28** of the second rotor **24** to the rotary shaft **18** via the female cone portion **27** and the male cone portion **25**, and also to reduce flux leakage from the attraction receiving portion **28** to the swash plate **23** via the female cone portion **27**. The reduction of the flux leakage inhibits the reduction of the electromagnetic force of the solenoid **32** acting on the attraction receiving portion **28**.

The following will describe the tenth embodiment of the present invention with reference to FIGS. **19** and **20**. The same reference numerals are used for the common elements or components in the first and tenth embodiments, and the description of such elements or components for the tenth embodiment will be omitted.

Referring to FIG. **20**, the surface **281** of the attraction receiving portion **28** of the second rotor **24** has a compression-stroke corresponding region **75** and a suction-stroke corresponding region **77**. Referring to FIGS. **19** and **20**, a part of the compression-stroke corresponding region **75** which is designated by **76** and a part of the suction-stroke corresponding region **77** which is designated by **78** cooperate to form a planar inclined portion **74** that is spaced from the solenoid **32** with a radially outwardly increasing spaced distance. The boundary between the part **76** of the compression-stroke corresponding region **75** and the part **78** of the suction-stroke corresponding region **77** is located at the top-dead-center corresponding position **79**. The compression-stroke corresponding region **75** is an angular range which is centered around the axis **181** and in which the axial centers **451** of the pistons **45** (only one piston being shown in FIG. **20**) in the compression stroke are present. The suction-stroke corresponding region **77** is an angular range which is centered around the axis **181** and in which the axial centers **451** of the pistons **45** in the suction stroke are present. The hinge mechanism **40** is located behind the inclined portion **74** of the second rotor **24**.

While the cone clutch K is in the disengaged state, there is fear that the second rotor **24** may be inclined in the direction that causes the upper side of the attraction receiving portion **28** of FIG. **19** to be moved toward the solenoid **32**. The presence of the inclined portion **74** of the attraction receiving portion **28** of the second rotor **24** helps to prevent the attraction receiving portion **28** of the second rotor **24** from being moved into harmful contact with the solenoid **32**.

The following will describe the eleventh embodiment of the present invention with reference to FIGS. **21** and **22**. The same reference numerals are used for the common elements or components in the first and eleventh embodiments, and the description of such elements or components for the eleventh embodiment will be omitted.

Referring to FIG. **21**, the reference symbol $T_{cmin}$ denotes the clearance between the top end of the piston **45** and the suction valve plate **15** that is formed when the swash plate **23** is at the maximum inclination angle position. The clearance will be referred to merely as "top clearance" of the piston **45**, hereinafter. The positions of the top end of the piston **45** and the attraction receiving portion **28** of the second rotor **24** when the swash plate **23** is at the maximum inclination angle position are indicated by the chain double-dashed line in FIG. **21**.

Referring to FIG. **22**, when the top clearance of the piston **45** that is formed when the swash plate **23** is at the minimum inclination angle position is represented by $\Delta T_c + T_{cmin}$, and the amount of elastic deformation of the disc spring **31** (spring member) when the cone clutch K is engaged is represented by $\eta$, $\Delta T_c$ is set so as to meet the relational expression $T_{cmin} + \Delta T_c \geq \eta$.

The horizontal axis of the graph represents the inclination angle $\theta$ of the swash plate **23** and the vertical axis of the graph represents the dimension of the top clearance of the piston **45**. The curve M shows a change of the top clearance.

If the top end of the piston **45** comes into contact with the suction valve plate **15** while the swash plate **23** is at the minimum inclination angle position, the spring force of the disc spring **31** produced when the swash plate **23** is at the minimum inclination angle position acts on the piston **45**. The reaction force of the spring force urges the second rotor **24** toward the first rotor **22**. This makes it easier for the second rotor **24** to rotate with the first rotor **22** thereby to increase the mechanical loss of the variable displacement compressor **10**.

By setting the $\Delta T_c$ as described above, however, the contact between the top end of the piston **45** and the suction valve plate **15** is avoided.

The following will describe the twelfth embodiment of the present invention with reference to FIGS. **23** and **24**. The same reference numerals are used for the common elements or components in the first and twelfth embodiments, and the description of such elements or components for the twelfth embodiment will be omitted.

Referring to FIGS. **23** and **24**, a pair of grooves **80** (only one groove being shown in FIG. **23**) is formed in the conical surface **271** of the second rotor **24** so as to extend linearly across the conical surface **271**. The pair of grooves **80** is formed at positions within an angular range **82** around the axis **181**. As shown in FIG. **24**, the angular range **82** covers an angular range around the axis **181** excepting the angular range $\gamma$ ranging from the top-dead-center corresponding position **79** to a position that is spaced at a predetermined angle $\gamma$ in the compression-stroke corresponding region **75**. The angular range $\gamma$ is, for example, 45°. When the swash plate **23** is being moved from the minimum inclination angle position, or when the cone clutch K is being shifted from the disengaged state to the engaged state, partial contact tends to occur between the conical surfaces **251** and **271** in the angular range $\gamma$. If any groove such as **80** is formed in the annular range $\gamma$, a wear tends to occur in the angular range $\gamma$. The angular range **82** within which the grooves **80** are formed is set for prevention of the wear.

The grooves **80** allow lubricating oil to flow smoothly into the gap between the conical surfaces **251** and **271**. The grooves **80** also serve any foreign matters present between the conical surfaces **251** and **271** to be caught. If grooves such as **80** are formed in the conical surface **251** of the first rotor **22**, there is fear that the foreign matters may be flown out of the grooves **80** into the gap between the conical surfaces **251** and **271** once again by the centrifugal force caused by the rotation of the first rotor **22**. In the present embodiment wherein the grooves **80** are formed in the conical surface **271**, however, such problem may be prevented.

The following will describe the thirteenth embodiment of the present invention with reference to FIGS. **25** and **26**. The same reference numerals are used for the common elements or components in the first and thirteenth embodiments, and the description of such elements or components for the thirteenth embodiment will be omitted.

Referring to FIG. **25**, a suction pressure sensor **84** and a discharge pressure sensor **85** are connected to the control computer C by signals. The suction pressure sensor **84** detects the pressure in the suction chamber **131** (or suction pressure), and the discharge pressure sensor **85** detects the pressure in the discharge chamber **132** (or discharge pressure). Data on the suction pressure detected by the suction pressure sensor **84** and data on the discharge pressure detected by the discharge pressure sensor **85** are transmitted by the respective sensors to the control computer C. The control computer C controls the energization and deenergization of the solenoid **32** based on the data on the suction pressure and the discharge pressure detected by the suction pressure sensor **84** and the discharge pressure sensor **85**, respectively.

FIG. **26** is a flowchart illustrating a control program that controls the energization and deenergization of the solenoid **32**. The control computer C executes the control program of

FIG. **26**. The following will describe the energization and deenergization control of the solenoid **32** based on the flowchart of FIG. **26**.

At step S**1**, the control computer C determines whether or not the displacement control valve **56** is ON. If the displacement control valve **56** is ON (YES at step S**1**), the control computer C energizes the solenoid **32** at step S**2** thereby to shift the cone clutch K from the disengaged state to the engaged state. At step S**3**, the control computer C determines whether or not the differential pressure ΔP (=Pd−Ps) between the discharge pressure Pd detected by the discharge pressure sensor **85** and the suction pressure Ps detected by the suction pressure sensor **84** is greater than or equal to a preset differential-pressure reference value z.

If the differential pressure ΔP does not reach the preset differential-pressure reference value z (NO at step S**3**), the control computer C continues the energization of the solenoid **32** at step S**2**. If the cone clutch K is engaged completely by the continuation of the energization of the solenoid **32**, the second rotor **24** and the swash plate **23** are rotated integrally with the first rotor **22**.

If the differential pressure ΔP reaches the preset differential-pressure reference value z (YES at step S**3**), the control computer C causes the solenoid **32** to be deenergized at step S**4**. If the differential pressure (=Pd−Ps) is relatively small when the swash plate **23** is at the minimum inclination angle position, the force of the swash plate **23** that presses the second rotor **24** against the first rotor **22** is also relatively small, which may cause the female cone portion **27** to slide on the male cone portion **25**. If the solenoid **32** is deenergized in such a state of the differential pressure ΔP, the rotation of the first rotor **22** is not steadily transmitted to the swash plate **23** via the second rotor **24**, so that the variable displacement compressor **10** fails to be started.

The differential-pressure reference value z is set so that the female cone portion **27** does not slide on the male cone portion **25**. Therefore, the variable displacement compressor **10** may be steadily started.

The following will describe the fourteenth embodiment of the present invention with reference to FIGS. **27** and **28**. The same reference numerals are used for the common elements or components in the thirteenth and fourteenth embodiments, and the description of such elements or components for the fourteenth embodiment will be omitted.

Referring to FIG. **27**, a speed sensor **89** for detecting the speed of a vehicle engine (not shown) is connected to the control computer C by signals. A temperature sensor **90** for detecting the temperature of the outside air near the evaporator **52** (or blow off temperature) is connected to the control computer C by signals. Data on the speed detected by the speed sensor **89** is sent to the control computer C. The control computer C calculates the change of the speed (or rotational acceleration) based on the data on the speed detected by the speed sensor **89**. The control computer C controls the energization and deenergization of the solenoid **32** based on the data of the speed and the discharge pressure detected by the speed sensor **89** and the discharge pressure sensor **85**, respectively.

FIG. **28** is a flowchart illustrating a control program that controls the energization and deenergization of the solenoid **32**. The control computer C executes the control program of FIG. **28**. The following will describe the energization and deenergization control of the solenoid **32** based on the flowchart of FIG. **28**.

At step S**11**, the control computer C determines whether or not the displacement control valve **56** is ON. If the displacement control valve **56** is ON (YES at step S**11**), the control computer C estimates at step S**12** the suction pressure from the duty ratio with which the passage of the electric current through the displacement control valve **56** is controlled and the temperature detected by the temperature sensor **90**. At step S**13**, the control computer C estimates the compression force from the estimated suction pressure and the discharge pressure detected by the discharge pressure sensor **85**.

At step S**14**, the control computer C estimates the transmission torque G from the estimated compression force. The transmission torque G refers to a value of the torque that is transmitted by the compression force through the cone clutch K. At step S**15**, the control computer C estimates the load torque H from the operating conditions (the speed and the rotational acceleration) of the variable displacement compressor **10**. The load torque H refers to a value of the torque that needs to be transmitted from the first rotor **22** to the second rotor **24** through the cone clutch K.

At step S**16**, the control computer C determines whether or not the transmission torque G is greater than or equal to the load torque H. If the transmission torque G does not reach the load torque H (NO at step S**16**), the control computer C energizes the solenoid **32**. The energization of the solenoid **32** increases the engagement force of the cone clutch K thereby to cause the second rotor **24** to be rotated integrally with the first rotor **22**.

When the displacement control valve **56** is ON and the swash plate **23** is located at a position close to the minimum inclination angle position, the variable displacement compressor **10** may be operated at the minimum displacement. Such operation of the compressor **10** occurs, for example, when the outside air temperature is extremely low. If the solenoid **32** is then in the deenergized state, there is fear that the torque of the first rotor **22** may not be transmitted to the second rotor **24**, that is, the second rotor **24** may not be rotated integrally with the first rotor **22**.

In the present embodiment, the integral rotation of the second rotor **24** with the first rotor **22** is steadily ensured while the variable displacement compressor **10** is operating at the minimum displacement.

The following will describe the fifteenth embodiment of the present invention with reference to FIG. **29**. The same reference numerals are used for the common elements or components in the first and fifteenth embodiments, and the description of such elements or components for the fifteenth embodiment will be omitted.

Referring to FIG. **29**, an annular permanent magnet **86** is fixedly mounted in the surface **281** of the attraction receiving portion **28** of the second rotor **24** that faces the solenoid **32**. The permanent magnet **86** receives the repulsive force from the solenoid **32** by passing electric current through the coil **33** of the solenoid **32** in the direction that is opposite to the direction of the electric current that causes the cone clutch K to be engaged. Thus, the cone clutch K may be shifted from the engaged state to the disengaged state.

The following will describe the sixteenth embodiment of the present invention with reference to FIG. **30**. The same reference numerals are used for the common elements or components in the first and sixteenth embodiments, and the description of such elements or components for the sixteenth embodiment will be omitted.

Referring to FIG. **30**, a first rotor **22**B corresponds to the first rotor **22** of the first embodiment and is made of a magnetic material. The first rotor **22**B is supported by the rotary shaft **18** in such a way that the first rotor **22**B is rotatable integrally with and slidable on the rotary shaft **18**. The first rotor **22**B has a male cone portion **25**B and an annular pressure receiving portion **26**B that extends radially outward from the outer periphery of the male cone portion **25**B. The male cone portion **25**B has a conical surface **251**B. The solenoid **32**B corresponds to the solenoid **32** of the first embodiment and is mounted in the front housing **12**. The solenoid **32**B attracts the male cone portion **25**B when an electric current is passed through the coil **33**.

The second rotor **24**B corresponds to the second rotor **24** of the first embodiment. The second rotor **24**B is fitted on and supported by the pressure receiving portion **26**B of the first rotor **22**B so as to be slidable and relatively rotatable on the first rotor **22**B. The second rotor **24**B has a female cone portion **27**B and a pair of projections **37** and **38** (only one projection **37** being shown in FIG. **30**). The pair of projections **37** and **38** forms a part of the hinge mechanism **40**. The female cone portion **27**B has a conical surface **271**B. The male cone portion **25**B and the female cone portion **27**B cooperate to form the cone clutch K.

The thrust bearing **30** and the disc spring **31** are interposed between the male cone portion **25**B of the first rotor **22**B and the female cone portion **27**B of the second rotor **24**B. The thrust bearing **29** is interposed between the first rotor **22**B and the front housing **12**. The reaction force developed when the refrigerant is discharged from the compression chamber **112** is received by the front housing **12** via the swash plate **23**, the second rotor **24**B, the cone clutch K, the first rotor **22**B and the thrust bearing **29**.

An annular stop **87** is mounted on the rotary shaft **18** at a position between the inclination-angle reduction spring **41** and the male cone portion **25**B of the first rotor **22**B for restricting the distance of the first rotor **22**B from the solenoid **32**B in the direction of the axis **181**.

The swash plate **23** has at a position adjacent to the hinge mechanism **40** a pressing arm **88** that extends toward the pressure receiving portion **26**B of the first rotor **22**B. The pressure receiving portion **26**B has a cam surface **261**. The end of the pressing arm **88** is in contact with the cam surface **261**. The pressing arm **88** is pressed against the cam surface **261** when the swash plate **23** is changed from the minimum inclination angle position to the maximum inclination angle position. The cam surface **261** plays the role of the cam surface **391** of the first embodiment.

When the solenoid **32**B is energized with the swash plate **23** located at the minimum inclination angle position, the solenoid **32**B attracts the first rotor **22**B thereby to shift the cone clutch K from the disengaged state to the engaged state. Thus, the rotation of the rotary shaft **18** is transmitted to the swash plate **23** via the first rotor **22**B, the cone clutch K, the second rotor **24**B and the hinge mechanism **40**.

The sixteenth embodiment of the present invention has substantially the same effects as those which are described under the items (1), (2), (4) and (7) of the first embodiment.

The following will describe the seventeenth embodiment of the present invention with reference to FIGS. **31** and **32**. The same reference numerals are used for the common elements or components in the second and seventeenth embodiments, and the description of such elements or components for the seventeenth embodiment will be omitted.

Referring to FIG. **31**, the disc spring **91** is interposed between the thrust bearing **30** and the second rotor **24**A at a position adjacent to the hinge mechanism **40**. The disc spring **91** is disposed in a recess **92** formed on the surface **272** of the second rotor **24**A and plays the role of the disc spring **31** of the first embodiment.

Referring to FIG. **32**, the disc spring **91** is positioned within an angular range α around the axis **181** that ranges between the top-dead-center corresponding position **79** and a position angularly spaced from the top-dead-center corresponding position **79** at a predetermined angle α in the compression-stroke corresponding region **75**. In the embodiment of FIG. **32**, the angular range α is 90°. The arrow F**6** of FIG. **31** denotes an imaginary spring load produced if the disc spring **31** of the second embodiment of FIG. **6** is used instead of the disc spring **91**. The arrow FL of FIG. **31** denotes the reaction force received by the swash plate **23** via the pistons **45**. The spring load F**6** of the disc spring **31** acts evenly on the compression-stroke corresponding region **75** and the suction-stroke corresponding region **77** of the second rotor **24**A.

The reaction force FL is larger in the compression-stroke corresponding region **75** than in the suction-stroke corresponding region **77**. That is, the reaction force FL acts on the second rotor **24**A eccentrically. Therefore, a moment FL×Lh is produced acting on the second rotor **24**A.

When the displacement control valve **56** (refer to FIG. **1**) is switched from ON state to OFF state, or when the cone clutch K is shifted from the engaged state to the disengaged state, the moment FL×Lh acting on the second rotor **24**A causes the second rotor **24**A to incline relative to the first rotor **22**A thereby to apply forces X**1** and X**2** to the second rotor **24**A. Thus, the second rotor **24**A moved when the displacement control valve **56** is switched from the ON state to the OFF state is subjected to friction force caused by the forces X**1** and X**2**. The friction force caused by the moment FL×Lh prevents the second rotor **24**A from moving smoothly, or prevents the cone clutch K from being shifted smoothly from the engaged state to the disengaged state.

In the present embodiment wherein the disc spring **91** is positioned within the angular range α of FIG. **32**, the spring load of the disc spring **91** prevents the inclination of the second rotor **24**A relative to the first rotor **22**A against the eccentric load of the reaction force FL thereby to allow the second rotor **24**A to move smoothly (or to allow the cone clutch K to be shifted smoothly from the engaged state to the disengaged state).

The following will describe the eighteenth embodiment of the present invention with reference to FIG. **33**. The same reference numerals are used for the common elements or components in the first and eighteenth embodiments, and the description of such elements or components for the eighteenth embodiment will be omitted.

Referring to FIG. **33**, the coil holder **34** has at the rear end thereof a radially inner annular end surface **34**A and a radially outer annular end surface **34**B. The annular end surface **34**A is closer to the second rotor **24** in the direction of the axis **181** than an outer peripheral surface **26**A of the pressure receiving portion **26** of the first rotor **22**. The coil holder **34** has on the radially inner annular portion thereof a first surface **341** that faces the outer peripheral surface **26**A of the pressure receiving portion **26** of the first rotor **22**. A first gap G**1** is formed between the outer peripheral surface **26**A of the pressure receiving portion **26** and the first surface **341** so as to form a path of magnetic flux that flows in the radial direction of the rotary shaft **18**.

The annular end surface **34**A is not located close to the second rotor **24**. A gap is formed between the annular end surface **34**A and the second rotor **24** for preventing magnetic flux from flowing in the direction of the axis **181** between the annular end surface **34**A and the second rotor **24**.

The annular end surface **34**B is located closer to the swash plate **23** in the direction of the axis **181** than the outer peripheral surface **28**A of the attraction receiving portion **28** of the second rotor **24**. The coil holder **34** has on the radially outer annular portion thereof a second surface **342** that faces the outer peripheral surface **28**A of the attraction receiving portion **28** of the second rotor **24**. A second gap G**2** is formed between the outer peripheral surface **28**A of the attraction receiving portion **28** and the second surface **342** of the coil holder **34** so as to form a path of magnetic flux that flows in the radial direction of the rotary shaft **18**. The disc spring **31** of the present embodiment is made of a non-magnetic material such as a stainless material.

When the solenoid **32** is energized, the magnetic flux developed in the coil holder **34** flows from the second surface **342** in the radial direction of the rotary shaft **18** to the outer peripheral surface **28**A of the attraction receiving portion **28** of the second rotor **24** via the second gap G**2**. The magnetic flux flowed to the second rotor **24** then flows to the first rotor **22** via a gap between the conical surfaces **251** and **271** of the first and second rotors **22** and **24**. The magnetic flux flowed to the first rotor **22** flows from the outer peripheral surface **26**A of the pressure receiving portion **26** in the radial direction of the rotary shaft **18** to the first surface **341** of the coil holder **34** via the first gap G**1**. That is, the magnetic flux developed in the coil holder **34** flows back to the coil holder **34** via the second gap G**2**, the second rotor **24**, the conical surfaces **251**, **271**, the first rotor **22** and the first gap G**1**, thus forming a magnetic circuit M**1**.

The magnetic flux that forms the magnetic circuit M**1** causes the conical surface **271** of the second rotor **24** to be attracted to the conical surface **251** of the first rotor **22** thereby to bring the conical surface **271** into contact with the conical surface **251**. In the present embodiment, the disc spring **31** that is made of a non-magnetic material prevents the magnetic flux from leaking from the second rotor **24** to the first rotor **22** via the thrust bearing **30** and the disc spring **31**, so that the magnetic flux flows through the gap between the conical surfaces **251** and **271**.

In order to adjust the gaps G**1** and G**2** which form a part of path of the magnetic flux flowing in the radial direction of the rotary shaft **18**, it is only necessary to adjust the radial length of any one of the coil holder **34**, the first rotor **22** and the second rotor **24**. Therefore, the required electromagnetic force of the solenoid **32** may be ensured by easy adjustment of the gaps G**1** and G**2**.

The following will describe the nineteenth embodiment of the present invention with reference to FIGS. **34** and **35**. The same reference numerals are used for the common elements or components in the third and nineteenth embodiments, and the description of such elements or components for the nineteenth embodiment will be omitted.

Referring to FIG. **34**, an annular pressing member **95** is interposed between the second rotor **24**A and the inclination-angle reduction spring **41** so as to surround the rotary shaft **18**. A gap is formed between the rotary shaft **18** and the inner peripheral surface of the pressing member **95**. The pressing member **95** is movable in the direction of the axis **181**. The pressing member **95** has a front surface **95**A that faces the second rotor **24**A and is in contact with the second rotor **24**A and the rolling bearing **62**. The pressing member **95** prevents the rolling bearing **62** from falling into the crank chamber **121**.

When the swash plate **23** is at the minimum inclination angle position and the cone clutch K is disengaged (or the solenoid **32** is deenergized), the inclination-angle reduction spring **41** does not urge the pressing member **95** toward the second rotor **24**A, but urges the swash plate **23** in the direction that causes the inclination angle of the swash plate **23** to be decreased. The front surface **95**A of the pressing member **95** is then spaced from the cylindrical guide portion **61**.

Referring to FIG. **35**, when the solenoid **32** is energized to engage the cone clutch K, the second rotor **24**A and the swash plate **23** are rotated integrally with the first rotor **22**A thereby to increase the inclination angle of the swash plate **23**. When the inclination angle of the swash plate **23** is increased, the swash plate **23** presses the inclination-angle reduction spring **41** against the pressing member **95** thereby to press the pressing member **95** against the second rotor **24**A. When the pressing member **95** presses the second rotor **24**A and the rolling bearing **62**, the second rotor **24**A is urged toward the first rotor **22**A thereby to increase the transmission torque between the conical surfaces **251** and **271**. In the present embodiment, the inclination-angle reduction spring **41** serves as an urging member. In addition, the inclination-angle reduction spring **41** and the pressing member **95** cooperate to form an urging device that urges the second rotor **24**A toward the first rotor **22**A. The front surface **95**A of the pressing member **95** is in contact with the second rotor **24**A. The inclination-angle reduction spring **41** and the pressing member **95** also serve as a distance restriction device that restricts the distance between the first rotor **22**A and the second rotor **24**A in the direction of the axis **181**.

The following will describe the twentieth embodiment of the present invention with reference to FIGS. **36** and **37**. The same reference numerals are used for the common elements or components in the third and twentieth embodiments, and the description of such elements or components for the twentieth embodiment will be omitted.

Referring to FIG. **36**, an annular pressing member **96** is interposed between the stop **42** and the inclination-angle reduction spring **41** so as to surround the rotary shaft **18**. A gap is formed between the rotary shaft **18** and the inner peripheral surface of the pressing member **96**. The pressing member **96** is movable in the direction of the axis **181**. The pressing member **96** has an annular end surface **96**A that faces the second rotor **24**A and is in contact with the second rotor **24**A. The stop **42** is disposed in the pressing member **96**.

Referring to FIG. **37**, when the solenoid **32** is energized to engage the cone clutch K, the second rotor **24**A and the swash plate **23** are rotated integrally with the first rotor **22**A thereby to increase the inclination angle of the swash plate **23**. With an increase of the inclination angle of the swash plate **23**, the inclination-angle reduction spring **41** is pressed against the pressing member **96** thereby to press the pressing member **96** against the second rotor **24**A. When the pressing member **96** presses the second rotor **24**A, the second rotor **24**A is urged toward the first rotor **22**A thereby to increase the transmission torque between the conical surfaces **251** and **271**. In the present embodiment, the inclination-angle reduction spring **41** serves as an urging member. In addition, the inclination-angle reduction spring **41** and the pressing member **96** cooperate to form an urging device that urges the second rotor **24**A toward the first rotor **22**A.

The following will describe the twenty-first embodiment of the present invention with reference to FIG. **38**. The same reference numerals are used for the common elements or components in the twentieth and twenty-first embodiments, and the description of such elements or components for the twenty-first embodiment will be omitted.

Referring to FIG. **38**, the first rotor **22**A has a female cone portion **27**C. The female cone portion **27**C has a conical surface **271**C that surrounds the axis **181**. The second rotor **24**A has a male cone portion **25**C that is connectable to and disconnectable from the female cone portion **27**C. The male cone portion **25**C has a conical surface **251**C that surrounds the axis **181**. The conical surfaces **271**C and **251**C are contactable in a face-to-face manner. Thus, the male cone portion **25**C of the second rotor **24**A and the female cone portion **27**C of the first rotor **22**A may cooperate to form the cone clutch K.

The present invention has been described in the context of the above embodiments, but it is not limited to those embodiments. It is obvious to those skilled in the art that the invention may be practiced in various manners as exemplified below.

A coil spring may be used as the spring member instead of the disc spring **31** interposed between the first rotor and the second rotor.

In the third embodiment, the guide may be interposed between the rotary shaft **18** and the second rotor **24**A as shown in FIG. **39**.

In the fifth embodiment, the first lubrication groove **65** and the second lubrication groove **66** may be formed only in the coil cover **64**.

In the sixth embodiment, the first annular lubrication groove **67** and the second annular lubrication groove **68** may be formed in the coil cover **64**. Although in the sixth embodiment the coil holder **34** has a single first annular lubrication groove **67** and a single second annular lubrication groove **68**, each of the coil cover **64** and the coil holder **34** may have a plurality of first annular lubrication grooves **67** and a plurality of second annular lubrication grooves **68**.

In the sixth embodiment, the first annular lubrication groove **67** may be dispensed with.

In the tenth embodiment, the whole surface **281** of the attraction receiving portion **28** that faces the solenoid **32** may be formed by an inclined surface such as the inclined portion **74**.

In the twelfth embodiment, either one of the paired grooves **80** may be dispensed with.

In the first embodiment, any wear-resistant surface treatment may be applied to the conical surfaces **251** and **271**.

A friction material may be used for at least one of the conical surfaces **251** and **271**. The use of the friction material improves the transmission of torque in the engaged cone clutch K.

Any member having a high wear resistance may be fitted on the male cone portion **25** thereby to form the conical surface **251**.

Any member having a high wear resistance may be fitted on the female cone portion **27** thereby to form the conical surface **271**.

The arms **35** and **36** of the swash plate **23** may be made of a non-magnetic material so as to prevent the magnetic flux from leaking from the attraction receiving portion **28** to the swash plate **23**.

In a modification of the thirteenth embodiment, a first discharge pressure, a first suction pressure or a first temperature (or blowoff temperature) of the outside air near the evaporator **52** when the swash plate **23** is at the minimum inclination angle position may be detected. Additionally, a second discharge pressure, a second suction pressure or a second temperature (or blowoff temperature) of the outside air near the evaporator **52** after the energization of the solenoid **32** is started may be detected. It may be so controlled that the solenoid **32** is deenergized when the value of change between the first discharge pressure and the second discharge pressure reaches a preset reference value. Alternatively, the solenoid **32** may be deenergized when the value of change between the first suction pressure and the second suction pressure reaches a preset reference value. Further alternatively, the solenoid **32** may be deenergized when the value of change between the first temperature and the second temperature reaches a preset reference value. The outside air temperature near the evaporator **52** is an element that reflects the pressure of refrigerant. The above-mentioned value of change of the discharge pressure, the suction pressure or the outside air temperature reflects the pressure differential between the discharge pressure and the suction pressure reasonably.

The male cone portion **25** may be made of a non-magnetic material.

In the seventeenth embodiment, the disc spring **91** may be positioned within an angular range β (<α) around the axis **181** that ranges between the top-dead-center corresponding position **79** and a position angularly spaced from the top-dead-center corresponding position **79** at a predetermined angle **13** in the suction-stroke corresponding region **77**.

In the seventeenth embodiment, a coil spring may be used instead of the disc spring **91**.

In the eighteenth embodiment, the disc spring **31** may be made of a magnetic material.

In the nineteenth and twentieth embodiments, the urging device including the inclination-angle reduction spring **41** and the pressing member **95** or **96** may be configured otherwise.

What is claimed is:

**1**. A swash plate type variable displacement compressor comprising:

a rotary shaft;

a swash plate rotated by driving force of the rotary shaft, the swash plate being inclinable at a variable inclination angle;

a plurality of pistons engaged with the swash plate, the pistons being reciprocable in accordance with the rotation of the swash plate so that a length of stroke of each piston is varied depending on the inclination angle of the swash plate;

a first rotor connected to the rotary shaft for rotation therewith;

a second rotor transmitting the rotation of the first rotor to the swash plate;

a solenoid producing electromagnetic force that acts on the first rotor or the second rotor so that the first rotor and the second rotor move toward each other; and

a cone clutch engageable by energization of the solenoid, the cone clutch having a male cone portion and a female cone portion, the male cone portion having a conical surface provided on one of the first rotor and the second rotor, the female cone portion having a conical surface provided on the other of the first rotor and the second rotor, the conical surface of the female cone portion being connectable to and disconnectable from the conical surface of the male cone portion.

**2**. The swash plate type variable displacement compressor according to claim **1**, wherein the second rotor is interposed between the swash plate and the first rotor, the conical surface of the male cone portion being provided on the first rotor, the conical surface of the female cone portion being provided on the second rotor, the solenoid being formed in an annular shape, the first rotor being located radially inward of the solenoid as viewed in a direction of an axis of the rotary shaft.

**3**. The swash plate type variable displacement compressor according to claim **1**, wherein a distance restriction device is provided for restricting a distance between the first rotor and the second rotor in a direction of an axis of the rotary shaft.

**4**. The swash plate type variable displacement compressor according to claim **3**, wherein an inclination-angle reduction spring is interposed between the first rotor and the swash plate for urging the swash plate in a direction that causes the inclination angle of the swash plate to be reduced, wherein the distance restriction device is a stop interposed between the inclination-angle reduction spring and the first rotor.

**5**. The swash plate type variable displacement compressor according to claim **4**, wherein the stop is a plain bearing.

6. The swash plate type variable displacement compressor according to claim 1, wherein the first rotor is formed in an annular shape to have an axial hole through which the rotary shaft is fixedly fitted, the conical surface of the male cone portion being provided on the first rotor.

7. The swash plate type variable displacement compressor according to claim 1, wherein a spacer is provided for spacing the conical surface of the male cone portion and the conical surface of the female cone portion from each other.

8. The swash plate type variable displacement compressor according to claim 7, wherein the spacer is a spring member interposed between the first rotor and the second rotor.

9. The swash plate type variable displacement compressor according to claim 8, wherein the spring member is a disc spring that surrounds an axis of the rotary shaft.

10. The swash plate type variable displacement compressor according to claim 8, wherein the spring member is disposed in a compression-stroke corresponding region.

11. The swash plate type variable displacement compressor according to claim 8, wherein a thrust bearing is interposed between the spring member and the second rotor or between the spring member and the first rotor.

12. The swash plate type variable displacement compressor according to claim 11, wherein the thrust bearing is a rolling bearing.

13. The swash plate type variable displacement compressor according to claim 8, wherein when a minimum top clearance of each piston that is formed when the swash plate is at a maximum inclination angle position is represented by $T_{cmin}$, a top clearance of each piston that is formed when the swash plate is at a minimum inclination angle position is represented by $\Delta T_c+T_{cmin}$, and an amount of elastic deformation of the spring member when the cone clutch is engaged is represented by $\eta$, $\Delta T_c$ meets the relational expression $T_{cmin}+\Delta T_c \geq \eta$.

14. The swash plate type variable displacement compressor according to claim 1, wherein a guide is interposed between the first rotor or the rotary shaft and the second rotor, the guide including a cylindrical guide portion and a cylindrical surface, the cylindrical guide portion being provided in one of the first rotor or the rotary shaft and the second rotor, the cylindrical surface being provided on the other of the first rotor or the rotary shaft and the second rotor, the cylindrical surface being rotatably and slidably fitted into or onto the cylindrical guide portion.

15. The swash plate type variable displacement compressor according to claim 14, wherein a radial bearing is interposed between the cylindrical guide portion and the cylindrical surface.

16. The swash plate type variable displacement compressor according to claim 15, wherein the radial bearing is a rolling bearing.

17. The swash plate type variable displacement compressor according to claim 1, wherein the solenoid has an annular surface that faces the second rotor, a lubrication groove being formed radially across the annular surface.

18. The swash plate type variable displacement compressor according to claim 17, wherein the lubrication groove includes a first lubrication groove below an axis of the rotary shaft, the first lubrication groove communicating at an outer periphery of the annular surface with a radially outer region of the solenoid, the first lubrication groove being located so as to be immersed in a lubricating oil accumulated in a crank chamber in which the swash plate is disposed.

19. The swash plate type variable displacement compressor according to claim 17, wherein the lubrication groove includes a second lubrication groove above an axis of the rotary shaft, the second lubrication groove communicating at an inner periphery of the annular surface with a radially inner region of the solenoid.

20. The swash plate type variable displacement compressor according to claim 1, wherein the solenoid has an annular surface that faces the second rotor, a lubrication groove being formed in the annular surface so as to extend along the annular surface.

21. The swash plate type variable displacement compressor according to claim 1, wherein the second rotor is made of a magnetic material and has an attraction receiving portion that is attracted to the solenoid by energization of the solenoid, the second rotor having a flux barrier that serves to reduce flux leakage from the attraction receiving portion to the rotary shaft or the swash plate.

22. The swash plate type variable displacement compressor according to claim 21, wherein the flux barrier is a void.

23. The swash plate type variable displacement compressor according to claim 1, wherein the swash plate is connected to the second rotor via a hinge mechanism at a position that is spaced radially from an axis of the rotary shaft, the second rotor having a surface that faces the solenoid, the surface of the second rotor having a inclined portion behind the hinge mechanism, the inclined portion being formed so as to be spaced from the solenoid with a radially outwardly increasing spaced distance.

24. The swash plate type variable displacement compressor according to claim 1, wherein the solenoid includes a coil and an annular coil holder that holds the coil, the coil holder having a radially outer annular end surface and a radially inner annular end surface that face the second rotor, at least one of the radially outer annular end surface and the radially inner annular end surface being tapered, the second rotor having an annular portion that faces the tapered surface of the coil holder, the annular portion of the second rotor having a surface that is tapered, the tapered surface of the annular portion of the second rotor being complementary to the tapered surface of the coil holder.

25. The swash plate type variable displacement compressor according to claim 1, wherein the solenoid includes a coil and an annular coil holder that holds the coil, the coil holder having on a radially inner annular portion thereof a first surface that faces an outer peripheral surface of the first rotor, the coil holder also having on a radially outer annular portion thereof a second surface that faces an outer peripheral surface of the second rotor, a first gap being formed between the outer peripheral surface of the first rotor and the first surface so as to form a path of magnetic flux that flows in a radial direction of the rotary shaft, a second gap being formed between the outer peripheral surface of the second rotor and the second surface so as to form a path of magnetic flux that flows in the radial direction of the rotary shaft, magnetic flux developed in the coil holder by energization of the solenoid being flowed back to the coil holder via the second gap, the second rotor, the conical surface of the male cone portion, the conical surface of the female cone portion, the first rotor and the first gap thereby to form a magnetic circuit.

26. The swash plate type variable displacement compressor according to claim 1, wherein a groove is formed in the conical surface of the female cone portion so as to extend across the conical surface of the female cone portion.

27. The swash plate type variable displacement compressor according to claim 26, wherein the groove is formed within an angular range around an axis of the rotary shaft, the angular range covering an angular range around the axis excepting the angular range ranging from a top-dead-center

corresponding position to a position that is spaced at a predetermined angle in a compression-stroke corresponding region.

**28**. The swash plate type variable displacement compressor according to claim **1**, wherein the solenoid and the cone clutch cooperate to form an electromagnetic clutch that is incorporated in a compressor housing of the swash plate type variable displacement compressor.

**29**. The swash plate type variable displacement compressor according to claim **1**, wherein an urging device is provided for urging the second rotor toward the first rotor with the cone clutch engaged by energization of the solenoid.

**30**. The swash plate type variable displacement compressor according to claim **29**, wherein the urging device includes a pressing member and an urging member, the pressing member being movable in a direction of an axis of the rotary shaft between the second rotor and the swash plate, the urging member being interposed between the pressing member and the swash plate, wherein when the inclination angle of the swash plate is increased, the swash plate presses the urging member against the pressing member thereby to urge the pressing member against the second rotor.

**31**. A method of controlling a solenoid of a swash plate type variable displacement compressor having a cone clutch engageable by energization of the solenoid, comprising the steps of:

starting passing an electric current through the solenoid;

detecting a differential pressure between a discharge pressure and a suction pressure after the step of starting passing the electric current through the solenoid; and

stopping passing the electric current through the solenoid if the differential pressure reaches a preset differential-pressure reference value.

**32**. A method of controlling a solenoid of a swash plate type variable displacement compressor having a swash plate and a cone clutch, the swash plate being inclinable at a variable inclination angle, the cone clutch being engageable by energization of the solenoid, comprising the steps of:

detecting a first pressure of a refrigerant or a first element that reflects the first pressure of the refrigerant when the swash plate is at a minimum inclination angle position;

starting passing an electric current through the solenoid;

detecting a second pressure of the refrigerant or a second element that reflects the second pressure of the refrigerant after the step of starting passing the electric current through the solenoid; and

stopping passing the electric current through the solenoid if a value of change between the first pressure and the second pressure reaches a preset reference value or if a value of change between the first element and the second element reaches a preset reference value.

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